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ASHAE-AIA Committee on Cooperation: John E. Haines, *Chairman*; C. B. Gamble, P. B. Gordon.

ASHAE-ASRE Committee on Cooperation: ASHAE Representatives—C. B. Gamble, L. N. Hunter, E. R. Queer. **ASRE** Representatives—Cecil Boling, A. J. Hess, H. F. Spoehrer.

Building: A. J. Offner, *Chairman*; P. B. Gordon, M. F. Blankin, M. W. Bishop, T. H. Urdahl.

Exposition: E. N. McDonnell, *Chairman*; John E. Haines, L. N. Hunter.

Honors and Awards: E. R. Queer, *Chairman*; John E. Haines, R. A. Sherman, B. H. Jennings, *Ex Officio*.

Long-Range Planning: John Everetts, Jr., L. N. Hunter (*three years*); John E. Haines†, E. L. Crosby (*two years*); W. A. Grant, J. Donald Kroeker (*one year*).

NRC—National Academy of Sciences, Div. of Engrg.: L. E. Seeley.

Nuclear Energy Engineering: R. A. Sherman, *Chairman*; W. F. Friend, W. A. Grant, L. N. Hunter, B. H. Jennings (Committee on Research).

Guide Advisory: C. E. Price, *Chairman*; R. E. Cherne, J. C. Fitts, P. B. Gordon.

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Nominating: J. Donald Kroeker, Portland, Ore., *Chairman*; B. L. Evans, St. Louis, *Secretary*; G. C. F. Asker, Falls Church, Va.; J. S. Burke, New Orleans; L. A. Childs, Syracuse; C. F. Hoffman, Toledo; B. J. Horsburgh, Montreal; Henry Kleinkauf, Omaha; B. M. Kluge, Milwaukee; G. A. Linskie, Dallas; H. B. Nottage, Santa Monica; W. B. Pennock, Ottawa; G. B. Supple, Indianapolis.

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† Temporary Chairman.

Local Chapter Officers—1956

Arizona

Organized 1953
Headquarters, Phoenix

President . . . . . J. B. Hoaglund
Vice President . . . . . E. P. Maxwell
Secretary . . . . . F. W. Gabbard
Treasurer . . . . . G. L. Jackson, Jr.
Board of Governors: W. A. Biddle, F. W. Gabbard, J. B. Hoaglund, G. L. Jackson, E. P. Maxwell

Arkansas

Organized 1952
Headquarters, Little Rock

President . . . . . J. W. Thompson
Vice President . . . . . H. H. Himstedt
Secretary . . . . . K. A. Pettit
Treasurer . . . . . W. A. Bullard
Board of Governors: J. L. Brown, W. A. Bullard, H. H. Himstedt, O. L. McCallister, Jr., L. B. Moon, K. A. Pettit, J. W. Thompson

Atlanta

Organized 1937
Headquarters, Atlanta, Ga.

President . . . . . F. W. Bull
Vice President . . . . . J. A. Marshall
Secretary . . . . . J. S. Edgar
Treasurer . . . . . W. P. West
Board of Governors: T. A. Barrow, Jr., J. M. Lazenby

Austin

Organized 1956
Headquarters, Austin, Texas

President . . . . . B. E. Segall, Jr.
Vice President . . . . . A. C. Lagow
Secretary . . . . . J. L. Rea, Jr.
Treasurer . . . . . P. R. Sisler
Board of Governors: D. F. Schuett, I. W. Wilke

Baltimore

Organized 1949
Headquarters, Baltimore, Md.

President . . . . . W. G. Robertson, Jr.
Vice President . . . . . R. E. Dressell
Secretary . . . . . C. E. McCaffray
Treasurer . . . . . W. C. Hossfeld
Board of Governors: G. M. Heslop, E. J. Morris, S. Y. Smith

Baton Rouge

Organized 1955
Headquarters, Baton Rouge, La.

President . . . . . J. F. Naylor, Jr.
Vice President . . . . . A. W. Holland
Secretary . . . . . W. J. LeBlanc
Treasurer . . . . . A. J. Mayers, Jr.
Board of Governors: G. S. Cox, H. J Weilbacher, C. S. Woodruff

Bluegrass

Organized 1954
Headquarters, Louisville, Ky.

President . . . . . W. J. Killian
Vice President . . . . . O. J. Hill
Secretary . . . . . A. E. Handmaker
Treasurer . . . . . H. L. Carr
Board of Governors: R. L. Craig, John Sutter, J. B. Uhl

British Columbia

Organized 1952
Headquarters, Vancouver, B. C., Canada

President . . . . . D. W. Thomson
Vice President . . . . . A. C. Martin
Secretary . . . . . W. F. Keenan
Treasurer . . . . . W. F. Wiggins
Board of Governors: D. B. Leaney, C. H. Turland

Central New York

Organized 1944
Headquarters, Syracuse

President . . . . . H. J. Morton
Vice President . . . . . Merle Weninger
Secretary . . . . . E. L. Moyer
Treasurer . . . . . S. F. Gilman
Board of Governors: R. A. Barr, D. J. Girard

Central Ohio

Organized 1944
Headquarters, Columbus

President . . . . . W. A. Schoonover
Vice President . . . . . T. R. Walker
Secretary . . . . . R. C. Liebert
Treasurer . . . . . J. W. Ford
Board of Governors: J. W. Ford, C. R. Graf, W. V. Jones, R. C. Liebert, W. A. Schoonover, T. R. Walker, R. A. Wilson

Cincinnati

Organized 1932
Headquarters, Cincinnati, Ohio

President . . . . . R. G. Anderson
Vice President . . . . . T. D. Reiley
Secretary . . . . . C. P. Krantz
Treasurer . . . . . E. A. Sobolewski
Board of Governors: W. H. Rieger

Connecticut

Organized 1940
Headquarters, New Haven

President . . . . . Walter Heywood
Vice President . . . . . Fritz Honerkamp
Secretary . . . . . R. B. Cahoon
Treasurer . . . . . J. D. Pierce
Board of Governors: E. J. Hoagland, J. O. Ostberg, F. J. Raible, Jr.

Local Chapter Officers—1956 (*Continued*)

Delta

Organized 1939
Headquarters, New Orleans, La.

President . . . . . H. N. Stall
Vice President . . . . . L. R. Maxwell
Secretary . . . . . O. F. Schully
Treasurer . . . . . R. K. Goode
Board of Governors: J. H. Maloney, E. H. Sanford

El Paso

Organized 1956
Headquarters, El Paso

President . . . . . J. O. LeBeau, Jr.
Vice President . . . . . G. H. Jackson
Secretary . . . . . J. H. Brooks
Treasurer . . . . . J. A. True
Board of Governors: W. A. Byars, Jr., W. S. Cooke, Jr., H. E. Taylor

Empire State Capital

Organized 1951
Headquarters, Albany, N. Y.

President . . . . . G. G. Davis
1st Vice President . . . . . H. F. Kruger
2nd Vice President . . . . . E. J. Mahoney
Secretary-Treasurer . . . . . M. E. Waddell
Board of Governors: H. S. Carleton, Jr., L. J. Mitsch, F. W. Sheldon

Golden Gate

Organized 1937
Headquarters, San Francisco, Calif.

President . . . . . F. K. Crouch
Vice President . . . . . Herb Duncan
Secretary . . . . . J. B. Smith
Treasurer . . . . . T. R. Simson
Board of Governors: R. C. Pribuss, J. I. Sprott

Illinois

Organized 1906
Headquarters, Chicago

President . . . . . H. G. Gragg
Vice President . . . . . H. E. Anderson
Secretary . . . . . Herbert Kreisman
Treasurer . . . . . E. P. Heckel, Jr.
Board of Governors: R. J. Salinger, L. H. Streb, E. R. Teske

Illinois-Iowa

Organized 1956
Headquarters, Moline

President . . . . . R. L. Graham
Vice President . . . . . W. W. Dodge
Secretary . . . . . D. H. Ninow
Treasurer . . . . . E. O. Hull
Board of Governors: W. L. Davis, C. P. Rohde, J. M. Hartman

Indiana

Organized 1943
Headquarters, Indianapolis

President . . . . . A. O. Roche, Jr.
Vice President . . . . . G. V. Vogel
Secretary . . . . . W. F. Currise
Treasurer . . . . . J. M. Teskoski
Board of Governors: J. W. Jackson, A. B. Keller, W. L. Kercheval

Inland Empire

Organized 1950
Headquarters, Spokane, Wash.

President . . . . . Max Tonn
Vice President . . . . . J. A. Doyle
Secretary . . . . . R. D. Nevers
Treasurer . . . . . J. R. Morris
Board of Governors: D. E. Arnold. R. O. Haneberg, F. W. Jenkinson

Iowa

Organized 1940
Headquarters, Des Moines

President . . . . . D. E. Schroeder
Vice President . . . . . W. A. Schworm
Secretary-Treasurer . . . . . R. T. Howard, Jr.
Board of Governors: H. E. Drain, G. J. Kraai, H. A. Schwieger

Kansas

Organized 1951
Headquarters, Wichita

President . . . . . H. M. Skalla
Vice President . . . . . T. L. Roberts
Secretary . . . . . J. C. Mikesell
Treasurer . . . . . E. L. Oswald
Board of Governors: A. B. Newton, R. L. Pennington, Charles Yoe

Kansas City

Organized 1917
Headquarters, Kansas City, Mo.

President . . . . . R. M. Spencer
Vice President . . . . . S. C. McCann
Secretary . . . . . J. E. Miller
Treasurer . . . . . J. C. Fasnacht
Board of Governors: J. R. DeRigne, J. S. Dukelow, T. I. Harriman, A. S. Hurt, Jr., A. A. Zahner

Manitoba

Organized 1935
Headquarters, Winnipeg, Man., Canada

President . . . . . A. H. Millar
Vice President . . . . . H. F. Randall
Secretary-Treasurer . . . . . R. M. Fraser
Board of Governors: W. L. Algie, G. T. Christie, G. C. Davis, W. D. Fraser, N. B. Jorgensen, J. J. MacKenzie, K. J. McCartney, A. K. Piercy, J. R. Stephenson, D. S. Swain

Local Chapter Officers—1956 (*Continued*)

Massachusetts

Organized 1912
Headquarters, Boston

President . . . . . W. G. Martin, Jr.
Vice President . . . . . E. L. Blair
Secretary . . . . . F. J. Butler
Treasurer . . . . . R. F. Curry
Board of Governors: F. J. Butler, G. Donald Fife, J. P. Hoar, J. A. Ivester, C. D. Potter, R. B. Stevens, E. M. Tracey, R. G. Vanderweil, H. F. White

Memphis

Organized 1944
Headquarters, Memphis, Tenn.

President . . . . . W. L. Henson
Vice President . . . . . R. H. Bolding
Secretary . . . . . G. B. Ramsey, Jr.
Treasurer . . . . . R. L. Elbrecht
Board of Governors: R. H. Bolding, R. L. Elbrecht, W. L. Henson, G. B. Ramsey

Miami Valley

Organized 1950
Headquarters, Dayton, Ohio

President . . . . . J. B. Rishel
Vice President . . . . . L. P. Brehm
Secretary . . . . . R. B. Walcott
Treasurer . . . . . N. O. Mitchell
Board of Governors: W. R. Budde, R. J. Perkins, J. M. Schweiger, D. E. Tullis, C. D. Weaver, Jr.

Michigan

Organized 1916
Headquarters, Detroit

President . . . . . J. N. Livermore
Vice President . . . . . P. S. Hosman
Secretary . . . . . J. B. Olivieri
Treasurer . . . . . K. J. Wagoner
Board of Governors: J. G. Black, Jr., D. S. Falk, Axel Marin, G. S. Whittaker

Minnesota

Organized 1918
Headquarters, Minneapolis

President . . . . . E. T. Erickson
Vice President . . . . . F. G. Vogt
Secretary . . . . . J. H. Jester
Treasurer . . . . . R. G. Gridley
Board of Governors: G. M. Brown, M. E. Quigley

Mississippi

Organized 1953
Headquarters, Jackson

President . . . . . B. L. Palmer
Vice President . . . . . J. W. Doggett, Jr.
Secretary . . . . . M. F. Knipp, Jr.
Treasurer . . . . . T. D. Luke
Board of Governors: W. H. Lambeth, W. H. Winders

Montreal

Organized 1936
Headquarters, Montreal, Que., Canada

President . . . . . S. R. Plamondon
Vice President . . . . . W. E. Jarvis
Secretary . . . . . Arthur DeBreyne
Treasurer . . . . . Paul Thibault
Board of Governors: W. C. Hole, B. J. Horsburgh, R. J. Ker, G. N. Martin, H. G. S. Murray

Nebraska

Organized 1940
Headquarters, Omaha

President . . . . . L. J. Paulsen
Vice President . . . . . M. F. Stober
Secretary . . . . . E. N. Seiler
Treasurer . . . . . F. P. Manchester, Jr.
Board of Governors: R. C. Dearing, R. N. Gates

New Mexico

Organized 1954
Headquarters, Albuquerque

President . . . . . C. R. Wherritt
Vice President . . . . . J. K. James
Secretary . . . . . R. G. Merryman
Treasurer . . . . . T. W. Eglinton
Board of Governors: F. H. Bridgers, Marcello Giomi

New York

Organized 1911
Headquarters, New York

President . . . . . J. B. Hewett
Vice President . . . . . C. R. Hiers
Secretary . . . . . C. H. Flink
Treasurer . . . . . W. M. Heebner
Board of Governors: L. D. Carr, Albert Giannini, W. O. Huebner, J. M. Levy, W. J. Olvany

North Jersey

Organized 1952
Headquarters, Newark, N. J.

President . . . . . W. C. Kruse, Jr.
Vice President . . . . . M. C. Christesen
Secretary . . . . . S. H. Nitzberg
Treasurer . . . . . L. G. Huggins
Board of Governors: H. P. Morehouse, C. E. Parmelee, C. H. Smith, C. W. Zimmer

North Texas

Organized 1938
Headquarters, Dallas

President . . . . . J. A. Ray
Vice President . . . . . H. G. Gregerson
Secretary . . . . . G. H. Meffert
Treasurer . . . . . Ross Zumwalt
Board of Governors: M. W. Brown, D. C. Kieffer, F. L. McFadden

Local Chapter Officers—1956 (*Continued*)

Northeastern Oklahoma

Organized 1948
Headquarters, Tulsa

President . . . . . J. T. McKinney
Vice President . . . . . J. D. Ryan
Secretary . . . . . G. A. McCune
Treasurer . . . . . H. W. Meinholtz
Board of Governors: H. L. Cauvel, J. R. Shipman, R. W. Winget

Northern Alberta

Organized 1956
Headquarters, Edmonton, Alta., Canada

President . . . . . W. C. Woods
Vice President . . . . . P. M. Butler
Secretary . . . . . E. K. Cumming
Treasurer . . . . . R. A. Williams
Board of Governors: Harry Hole, H. A. G. Maybank, A. N. Nordstrom, D. A. Peckinpaugh, R. G. Proudfoot, Kornelius Siemens

Northern Ohio

Organized 1916
Headquarters, Cleveland

President . . . . . H. R. Canoyer
Vice President . . . . . R. E. Leising
Secretary . . . . . R. D. Wilson
Treasurer . . . . . R. A. Urban
Board of Governors: L. C. Burkes, J. L. Frisse, Sherwood Nassau

Northern Piedmont

Organized 1952
Headquarters, Greensboro, N. C.

President . . . . . L. P. Gorrell
Vice President . . . . . C. V. Creefer
Secretary . . . . . C. E. Amick
Treasurer . . . . . E. A. Stroupe, Jr.
Board of Governors: R. B. Knight, S. T. Oliver, M. G. Saunders

Oklahoma

Organized 1935
Headquarters, Oklahoma City

President . . . . . F. R. Denham
Vice President . . . . . A. C. Shelley
Secretary . . . . . W. R. Johnson
Treasurer . . . . . W. H. Stewart
Board of Governors: W. J. Collins, Jr., E. F. Dawson, J. R. Patten

Ontario

Organized 1922
Headquarters, Toronto, Ont., Canada

President . . . . . C. J. Bootes
Vice President . . . . . Charles Torry
Secretary-Treasurer . . . . . H. R. Roth
Board of Governors: D. L. Angus, Ernest Fox, R. A. Ritchie, Jack Thompson

Oregon

Organized 1939
Headquarters, Portland

President . . . . . W. R. Norte
Vice President . . . . . W. B. Hayes
Secretary . . . . . G. G. Van Alst
Treasurer . . . . . T. H. McClung
Board of Governors: Dick Blankenship, Keith Kruchek, R. L. Propst

Ottawa Valley

Organized 1952
Headquarters, Ottawa, Ont., Canada

President . . . . . D. W. Banton
Vice President . . . . . C. W. Watson
Secretary . . . . . Jacob Klassen
Treasurer . . . . . A. H. Hargreaves
Board of Governors: G. A. Gray, A. S. Lawson, E. J. Schoenherr, G. A. Wild

Pacific Northwest

Organized 1928
Headquarters, Seattle, Wash.

President . . . . . L. F. Christofferson
1st Vice President . . . . . E. W. Triol
2nd Vice President . . . . . M. W. McKinstry
Secretary . . . . . H. M. Hendrickson
Treasurer . . . . . R. H. Liniger
Board of Governors: T. C. Caskey, K. G. Massart

Philadelphia

Organized 1916
Headquarters, Philadelphia, Pa.

President . . . . . C. J. Lubking
1st Vice President . . . . . P. H. Yeomans
2nd Vice President . . . . . C. J. Forve
Secretary . . . . . Ludwig Mack
Treasurer . . . . . J. J. Hucker
Board of Governors: D. S. Plewes

Pittsburgh

Organized 1919
Headquarters, Pittsburgh, Pa.

President . . . . . R. M. Toucey
Vice President . . . . . E. H. Riesmeyer, Jr.
Secretary . . . . . G. E. Smetak
Treasurer . . . . . R. C. Firsching
Board of Governors: C. L. Benn, E. J. Busch, Stanley Stake

Rocky Mountain

Organized 1944
Headquarters, Denver, Colo.

President . . . . . A. S. Widdowfield
Vice President . . . . . D. D. Pearsall
Secretary . . . . . R. T. Beck
Treasurer . . . . . C. DiMercurio
Board of Governors: J. W. Braak, W. V. Burbank, L. L. DeLong

Local Chapter Officers—1956 (*Continued*)

Sacramento Valley

Organized 1952
Headquarters, Sacramento, Calif.

President.....V. W. Thornburg
Vice President.....M. J. Delavan
Secretary.....S. W. Doolittle
Treasurer.....J. A. White
Board of Governors: R. T. Andrews, W. B. Lander, R. A. Sarro

St. Louis

Organized 1918
Headquarters, St. Louis, Mo.

President.....W. P. Norris
Vice President.....N. J. Hubbuch
Secretary.....K. O. Williams
Treasurer.....J. J. Blackmore
oard of Governors: E. T. Clucas, A. W. Felt, W. C. Kaber, E. C. Kuntz, J. I. Levenhagen, C. K. Meyers

Shreveport

Organized 1948
Headquarters, Shreveport, La.

President.....H. E. Scott
Vice President.....A. L. Jones, Jr.
Secretary.....L. E. Kneipp
Treasurer.....J. L. Collins
Board of Governors: J. L. Collins, A. L. Jones, Jr., L. E. Kneipp, H. E. Scott, T. F. Scott, R. S. Segall, R. F. Zimmerman

South Carolina

Organized 1954
Headquarters, Columbia, S. C.

President.....J. E. McMurray
Vice President.....B. A. Leppard
Secretary.....A. L. Blackstone, Jr.
Treasurer.....H. J. Haar, Jr.
Board of Governors: H. W. Hood, Harry McKeown, R. K. Rouse

South Texas

Organized 1938
Headquarters, Houston

President.....J. C. Lewis
Vice President.....A. B. Ullrich, Jr.
Secretary.....E. B. Appling
Treasurer.....W. A. Sharp
Board of Governors: H. D. McMillan, Jr., J. H. Stephenson

Southern California

Organized 1930
Headquarters, Los Angeles

President.....H. F. Ulovec
Vice President.....W. J. Biggar
Secretary.....A. Z. Levine
Treasurer.....R. W. Thomas
Board of Governors: B. L. Hutchinson, Jr., G. D. Lord, R. C. Taylor, Albert Zimmerman

Southern Piedmont

Organized 1952
Headquarters, Charlotte, N. C.

President.....J. A. Rice
Vice President.....G. C. Garrett
Secretary.....E. D. Streng
Treasurer.....G. N. Payne, Jr.
Board of Governors: L. F. Lawrence, G. N. Payne, Jr., C. E. Petty

Southwest Texas

Organized 1946
Headquarters, San Antonio

President.....Boone Crisp
Vice President.....C. L. Herndon
Secretary.....D. T. Kern
Treasurer.....L. A. Ingle
Board of Governors: E. E. Cravens, W. H. Peck, Jr., Martin Royer

Toledo

Organized 1954
Headquarters, Toledo, Ohio

President.....J. F. Guest
Vice President.....J. M. Groskopf
Secretary.....Robert Greenwald
Treasurer.....D. L. Wilson
Board of Governors: R. O. Benington, Jr., G. A. Kelley, F. C. Richardson, Jr.

Utah

Organized 1944
Headquarters, Salt Lake City

President.....W. L. Stuewe
Vice President.....R. V. Oliver
Secretary-Treasurer.....W. C. Leslie
Board of Governors: J. L. Ashton*, A. R. Curtis, R. C. Evans, A. A. Maycock, D. R. Wilde

Virginia

Organized 1946
Headquarters, Norfolk

President.....C. G. Conaway, Jr.
Vice President.....D. C. Delinger
Secretary.....E. R. Simpson, Jr.
Treasurer.....W. L. Gibson
Board of Governors: A. B. Miller, A. R. Thompson, Jr.

Washington, D. C.

Organized 1935
Headquarters, Washington, D. C.

President.....J. H. Broome
Vice President.....G. W. Campbell
Secretary.....J. W. Morgan
Treasurer.....J. S. King
Board of Governors: J. J. Chaconas, Sr., B. P. Evans, W. C. Reamy

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* Filled unexpired term.

Local Chapter Officers—1956 (*Continued*)

West Texas

Organized 1953
Headquarters, Lubbock

President . . . . . J. C. Wharton
Vice President . . . . . H. L. Mayes
Secretary . . . . . D. G. Halley
Treasurer . . . . . V. B. Garry
Board of Governors: G. S. Cathey, R. A. Mixon, R. T. Roberts, T. F. Sartor

Western Massachusetts

Organized 1955
Headquarters, Springfield

President . . . . . J. E. Reed
Vice President . . . . . F. A. Ferraro
Secretary . . . . . C. R. Munn
Treasurer . . . . . J. J. Curran
Board of Governors: R. E. Cross, J. F. Hayes, A. M. Lovenberg, K. W. Maki, B. R. Washburn

Western Michigan

Organized 1931
Headquarters, Grand Rapids

President . . . . . F. L. Murray
Vice President . . . . . B. J. Walter
Secretary . . . . . R. J. Waalkes
Treasurer . . . . . K. L. Lamm
Board of Governors: R. R. Kingscott, S. M. Paganelli, William Wessels

Western New York

Organized 1919
Headquarters, Buffalo

President . . . . . V. N. Harwood
1st Vice President . . . . . G. E. Kuhn
2nd Vice President . . . . . J. P. Guerra
Secretary . . . . . J. F. Bedard
Treasurer . . . . . M. C. Beman
Board of Governors: M. C. Beman, Joseph Davis, Roswell Farnham, C. W. Kaupp, C. W. Stone

Wisconsin

Organized 1922
Headquarters, Milwaukee

President . . . . . H. K. Forfar
Vice President . . . . . H. W. Alyea
Secretary . . . . . H. W. Snyder
Treasurer . . . . . R. D. Rodwell
Board of Governors: F. W. Goldsmith, O. J. Ress, I. J. Rossiter

Special Branch

Switzerland

Organized 1952
Headquarters, Zurich

President . . . . . Walter Hausler
Vice President . . . . . John Frei
Secretary . . . . . L. F de Bruyn
Treasurer . . . . . Werner Niederer
Controller . . . . . Werner Gysi

Student Branches

North Carolina State College

Organized 1948
Headquarters, Raleigh

President . . . . . P. K. Tuggle
Vice President . . . . . J. G. Daye
Secretary . . . . . G. L. Waitman
Treasurer . . . . . W. B. Cavinese, Jr.
Reporter . . . . . M. K. Sloan
Engineer's Council . . . . . R. J. Parker

Oregon State College

Organized 1949
Headquarters, Corvallis

President . . . . . P. I. Anderson
Secretary . . . . . L. D. Frost
Treasurer . . . . . R. G. Manfull

Purdue University

Organized 1949
Headquarters, W. Lafayette, Ind.

President . . . . . R. F. Puncochar
Secretary . . . . . G. T. Vail, Jr.
Co-ordinator . . . . . Richard Schultz

Texas A. & M. College

Organized 1946
Headquarters, College Station

President . . . . . J. W. Parsley
Vice President . . . . . W. W. Stubberman, Jr.
Secretary-Treasurer . . . . . T. H. Parish

University of Detroit

Organized 1949
Headquarters, Detroit, Mich.

President . . . . . R. J. Holtgreive
Vice President . . . . . D. F. Brennan
Secretary-Treasurer . . . . . C. N. Rollinger
Reporter . . . . . P. X. Mooney

University of Toronto

Organized 1951
Headquarters, Toronto, Ont., Canada

President . . . . . C. F. Smale
Secretary . . . . . T. D. Digel

1555

TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS

No. 1555

SIXTY-SECOND ANNUAL MEETING, 1956

CINCINNATI, OHIO

THE 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS featured a total of 8 technical sessions, January 23–25, 1956, with headquarters at the Sheraton-Gibson Hotel, Cincinnati, Ohio. Concurrent sessions were held each day. Attendance figures included: Members 541; Guests 144; Ladies 157.

The program (see page 19) included an opening luncheon address by F. W. Giesel, Business Manager, Cincinnati Post; a Kentucky Party on the evening of January 23rd; inspection trips to Cincinnati industries; a ladies theater party and a style show. Retiring Pres. John E. Haines was presented with the Past-President's Medal by Past-Pres. L. N. Hunter. The fourteenth award of the F. Paul Anderson Medal for 1955 was made to Dr. R. W. Keeton for his achievements and contributions through his studies of atmospheric environment and its effects upon the comfort and health of people, and his interest in maintaining scientific liaison between medical and engineering groups.

FIRST SESSION, MONDAY, JANUARY 23, 10:00 A.M.

After opening the first session in the Ballroom, President Haines announced that the Committee on Resolutions would be made up of John Everetts, Jr., Philadelphia, Pa., R. A. Sherman, Columbus, Ohio, and W. J. Collins, Jr., Oklahoma City, Okla.

President Haines then presented his annual report in which he reviewed many of the principal events of 1955. He also outlined the nature of some of the problems which arose and which received attention from the Officers and Council. For the first time the membership grew to exceed 10,000 and the number of chapters reached a total of 61. The President traveled more than 123,000 miles on Society business. The Council approved By-Law changes which, among other objectives, would result in a Regional Plan for Chapter Operations. The Council also approved Society membership in Engineers Joint Council. He felt that much progress was made during 1955, but that there is still much to be done by the incoming president and officers. (Full text of the Report of the President appears on pages 169 to 171, JOURNAL SECTION, *Heating, Piping & Air Conditioning*, March 1956).

REPORT OF TREASURER

President Haines then announced that Treas. E. R. Queer, University Park, Pa., would present the Treasurer's Report. Professor Queer had analyzed some important budget figures on a per member basis which he reported. The Accountant's Report which follows is a part of the Treasurer's report.

Accountant's Report

FRANK G. TUSA & CO.

CERTIFIED PUBLIC ACCOUNTANTS

37 Wall St., New York 5, N. Y.

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.
62 Worth Street, New York, N. Y.

Gentlemen:

Pursuant to your request, we have completed our examination of the books of account, the financial data pertinent thereto, and the records of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.—New York, N. Y. and Cleveland and its related funds for the fiscal year ended October 31, 1955 and submit herewith our report.

The audit covered a verification of the Assets and Liabilities as of the close of business October 31, 1955. For the fiscal year then ended, the recorded cash receipts were traced into the depositories; the cancelled bank checks were inspected and compared with the record of cash disbursements, and the disbursements were supported by payment vouchers. The dues from members and the interest from investments were accounted for.

A Balance Sheet reflecting the financial position of the Society as of the close of business October 31, 1955 is submitted herewith and your attention is directed to the following comments thereon:

CASH

The Cash on Deposit was verified by direct communication with commercial and savings banks and the balances reported to us were reconciled with those reflected by the books of the Society. A schedule of cash is included as a part of this report.

Checks representing the cash on hand for deposit were inspected by us and the cash on hand was counted.

MARKETABLE SECURITIES

The securities shown on the subjoined schedule were verified by direct communication with the Bankers Trust Company, where same are deposited for safe-keeping. Securities have been included in the accompanying balance sheet at the cost of acquisition plus the accumulated and accrued interest earned thereon.

ACCOUNTS RECEIVABLE

Trial balances taken of the membership dues receivable and sundry debtors as of the close of business October 31, 1955 were classified and aged as follows:

MEMBERSHIP	1955
Members	$ 4,384.63
Associates	3,073.00
Affiliates	4,868.95
Juniors	2,087.01
Students	7.00
TOTAL	$14,420.59

SUNDRY DEBTORS	
Unpaid charges made during 1955	$ 7,163.97

After writing off all receivables known to be uncollectible, it is our opinion that reserves for dues and accounts receivable doubtful of collection reflected in the accompanying Balance Sheet are ample to cover collection losses that may be incurred.

RESEARCH FUND ACCOUNTS RECEIVABLE

The balance due to the Research Fund from the Society as at October 31, 1955 represents the unpaid balance of 40 percent of the dues receivable from members, associates and affiliates as of such date.

INVENTORIES

The following inventories were verified either by physical count or direct communication.

TRANSACTIONS	$ 8,829.97
EMBLEMS	1,565.24
GUIDE PAPER	386.49
TOTAL	$10,781.70

A schedule of TRANSACTIONS inventories follows:

Volume	Year	Quantity	Price	Amount
1–54	1895–1948	4,866	$ .4000	$1,946.40
55	1949	262	2.0200	529.24
56	1950	679	2.0800	1,412.32
57	1951	695	2.2750	1,581.12
58	1952	112	1.7960	201.15
59	1953	648	2.0060	1,299.88
60	1954	900	2.0665	1,859.86
Total				$8,829.97

DEPOSITS RECEIVABLE

The deposit placed with the United Airlines in the sum of $425.00 was verified by direct communication.

ADVANCES

The indebtedness from employees in New York and Cleveland represents advances to the employees retirement plan in the sums of $3,643.46 and $1,339.77 respectively.

DEFERRED CHARGES

The following prepayments have been deferred to future operations:

HPAC—Subscriptions	$2,982.68
Insurance Premiums—Cleveland	3,503.49
Insurance Premiums—New York	260.32
TOTAL	$6,746.49

PERMANENT ASSETS

The Land and Buildings, Instruments, Equipment and Furniture and Fixtures are reflected on the Balance Sheet at cost of acquisition. With the exception of land and buildings all assets have been depreciated at the rate of 10 percent per annum. During the year fully depreciated Furniture and Fixtures in the amount of $324.47 was written off against the reserve.

In accordance with the resolution adopted by the Council at its meeting of January 23, 1949 depreciation was not provided on the buildings for the current fiscal year.

DEFERRED INCOME

The prepaid dues and initiation fees by members and candidates for membership have been deferred to future operations. The membership classification of the dues prepaid by elected members follow:

MEMBERSHIP	AMOUNT
Members	$23,915.68
Associates	8,362.15
Affiliates	1,697.50
Juniors	1,090.07
Students	19.50
TOTAL	$35,084.90

RESERVE FOR TRANSACTIONS

The unexpended balance of $13,962.43 as reflected at October 31, 1954 has been completely applied to the cost of TRANSACTIONS Volume 59 printed during the year.

TRANSACTIONS Volume 60 was also printed during the year so there was no necessity for any Reserve as at the end of the current fiscal year.

RESERVE FOR BUILDING MAINTENANCE

On October 31, 1952 the Council voted:

"That disbursements be made from the Building and Maintenance Reserve Fund as authorized by the Building Committee for major items of repair and maintenance, but any balance remaining in the fund beyond the expenditures for each year would go into reserve until the Building Maintenance Reserve Fund reached a total of $10,000 00."

The Fund had a balance at the beginning of the current fiscal year of $3,738.00. An additional $2,000.00 was provided in the budget for the current fiscal year. No major repairs chargeable to this Fund were made during the year.

FUNDS

There is included as a part of this report an analysis of Funds reflecting the changes that occurred therein during the fiscal year ended October 31, 1955.

INSURANCE

The insurance coverage of the Society follows:

FIRE		
Building (Cleveland)		$201,000.00
Personal Property:		
Cleveland	$104,000.00	
Printers	15,000.00	
New York	25,500.00	144,500.00
SPRINKLER LEAKAGE		
Waverly Press	10,000.00	
Hawes and Petit	2,000.00	12,000.00
NON-OWNERSHIP—AUTOMOBILE—COMPREHENSIVE		
Public Liability		100/300M
Property Damage		25M
Medical		1M

GENERAL PUBLIC LIABILITY—PREMISES

New York and Cleveland	
Liability	100/300M
Bodily Injury	25/100M

COMMERCIAL BLANKET BOND

All employees including the President, Treasurer and Chairman of the Finance Committee	$ 20,000.00

Respectfully submitted,

FRANK G. TUSA & CO., CERTIFIED PUBLIC ACCOUNTANTS

BALANCE SHEET—October 31, 1955

ASSETS

GENERAL FUND			
Society Assets (Per Schedule)			$ 68,293.18
Research Assets (Per Schedule)			81,594.39
PROPERTY FUND			
Land and Buildings	$84,920.96		
Less: Reserve for Depreciation	3,908.07	$81,012.89	
Laboratory and Equipment	76,726.43		
Less: Reserve for Depreciation	47,360.47	29,365.96	
Furniture and Fixtures (N.Y.)	19,190.98		
Less: Reserve for Depreciation	7,944.19	11,246.79	
Library (Cleveland)		300.00	
Tools (Cleveland)		300.00	122,225.64
BUILDING FUND			
Cash on Deposit		1,024.57	
Securities			
At Cost	37,164.00		
Add: Accumulated Interest	982.41	38,146.41	39,170.98
BUILDING MAINTENANCE RESERVE FUND			
Cash on Deposit			5,844.99
SOCIETY RESERVE FUND			
Cash on Deposit		27,656.14	
SECURITIES			
At Cost	88,166.50		
Add: Accumulated Interest	2,103.52	90,270.02	117,926.16
ENDOWMENT FUNDS			
F. Paul Anderson Fund			
Cash on Deposit		289.85	
Securities At Cost (Market Value $986.00)	1,000.00		
Add: Accumulated Interest	12.50	1,012.50	1,302.35
Research Fund			
Cash on Deposit			760.57
Homer Addams Fund			
Cash on Deposit			10,084.17
RESEARCH RESERVE FUND			
Cash on Deposit			52,368.87
TOTAL ASSETS			$499,571.30

LIABILITIES

GENERAL FUND				
Federal Withholding Taxes			$ 1,410.59	
Due Research General Fund			4,642.16	
Accrued Accounts				
Salaries and Commissions		$12,304.90		
Professional Fees		1,250.00	13,554.90	
Sundry Account Payable			100.00	
Deferred Income				
Prepaid Membership Dues				
Elected Members	$21,494.77			
Candidates	861.50	22,356.27		
Prepaid Admissions Fees		2,260.00	24,616.27	
RESERVE FOR FLUCTUATION IN CANADIAN EXCHANGE			9.40	$44,333.32
RESEARCH FUND				
Sundry Accounts Payable			138.17	
Deferred Income				
Unexpended Earmarked Contributions		19,318.98		
Prepaid Dues		13,590.13	32,909.11	33,047.28

NET WORTH

Society General Fund	23,959.86	
Research General Fund	48,547.11	
Property Fund	122,225.64	
Building Fund	39,170.98	
Building Maintenance Reserve Fund	5,844.99	
Society Reserve Fund	117,926.16	
F. Paul Anderson Award Fund	1,302.35	
Research Endowment Fund	760.57	
Research Reserve Fund	52,368.87	
Homer Addams Fund	10,084.17	422,190.70
TOTAL LIABILITIES AND NET WORTH		$499,571.30

COMPARATIVE STATEMENT OF INCOME AND EXPENSES

	FISCAL YEAR ENDED		
	OCTOBER 31 1955	OCTOBER 31 1954	INCREASE (DECREASE)
INCOME			
INCOME FROM DUES	$140,413.30	$133,756.91	$ 6,656.39
ADMISSION FEES	8,712.65	8,429.50	283.15
EMBLEMS, PINS, ETC.	1,558.75	819.75	739.00
PUBLICATIONS			
JOURNAL Contracts	23,500.00	23,500.00	–0–
THE GUIDE	163,552.46	144,845.24	18,707.22
TRANSACTIONS	1,358.08	1,473.83	(115.75)
Books, Reprints, Codes, Etc.	4,431.74	2,026.57	2,405.17
Membership Roll	7,685.50	–0–	7,685.50
INCOME FROM INVESTMENTS	908.55	1,282.89	(374.34)

RESEARCH			
Dues	73,967.74	69,784.26	4,183.48
U. S. Navy Research	1,599.68	13,340.89	(11,741.21)
Contributions—General	24,319.71	19,393.32	4,926.39
Contributions—Earmarked	82,934.45	41,079.96	41,854.49
Interest	502.78	43.87	458.91
TOTAL INCOME	*$535,445.39	$459,776.99	$75,668.40
EXPENSES			
COMMITTEES AND CHAPTERS	$ 29,869.95	$ 24,818.44	$ 5,051.51
MEETINGS	6,884.10	5,881.86	1,002.24
PUBLICATIONS			
Member Subscriptions to *HPAC*	20,042.06	18,139.37	1,902.69
TRANSACTIONS			
Volume 59	5,968.73	–0–	5,968.73
Volume 60	20,035.30	17,699.65	2,335.65
Membership Roll	5,796.67	2,236.22	3,560.45
Books, Reprints, Codes, Etc.	2,604.57	2,557.58	46.99
THE GUIDE	98,031.01	87,217.15	10,813.86
HEADQUARTERS	170,013.83	143,375.21	26,638.62
FUND RAISING	16,936.79	10,780.01	6,156.78
RESEARCH			
Committee Expenses	5,168.06	4,915.63	252.43
Staff Salaries	100,175.04	76,849.90	23,325.14
Laboratory Expenses	23,791.87	20,951.41	2,840.46
Building Operation and Maintenance	9,804.23	9,463.39	340.84
Provision for Building Maintenance Reserve	2,000.00	2,000.00	–0–
U. S. Navy Research	676.51	9,851.30	(9,174.79)
Cooperative Research	44,821.70	33,165.59	11,656.11
OTHER DEDUCTIONS	212.85	5,261.43	(5,048.58)
TOTAL EXPENSES	562,833.27	475,164.14	87,669.13
EXCESS OF EXPENSES OVER INCOME	*$ 27,387.88	$ 15,387.15	$12,000.73

* Exposition contributions of $63,449.83 for 1955 not included in this comparative statement.

The Report of Council was then presented by A. V. Hutchinson, New York, N. Y., Executive Secretary.

REPORT OF COUNCIL

During the calendar year, 1955 4 regular meetings of the Council were held, the organization meeting being convened on January 27, 1955. The President announced the personnel of the Council, General and Special Committees authorized by the By-Laws.

The Society has now completed one year of operations under its new name.

Council authorized membership in the *Engineers' Joint Council* effective in 1956.

The new Journal subscription plan became effective January 1, 1956; a new editorial contract was authorized and signed.

New By-Laws on membership grades to conform to *ECPD* recommendations were submitted to the membership in December 1955, also, proposed changes in chapter operations and Nominating Committee procedure.

As required by Section 46 of the Membership Corporations' Law of the State of New York the following report is presented by the Council and filed herewith:

Assets—October 31, 1955

The Society has the following Assets:	
Land and Laboratory Buildings at Cleveland, Ohio	$ 81,012.89
Furniture and Equipment, Tools, etc., at New York and Cleveland	41,212.75
On deposit in New York and Cleveland banks	202,804.18
Securities (United States and Utility Bonds) with Bankers Trust Co., New York as custodian	133,238.79
Accounts Receivable	18,366.27
Inventory Items	10,781.70
Deferred Items	6,746.49
Miscellaneous	766.07
TOTAL ASSETS—1955	$494,929.14
TOTAL ASSETS—1954	422,639.70
Increase for the year	$ 72,289.44

Liabilities—October 31, 1955

The Society has the following Liabilities:	
Accounts Payable	$ 238.17
Federal Withholding Taxes	1,410.59
Accrued Accounts	13,554.90
Deferred Income:	
Research Projects	19,318.98
Prepaid Dues, Initiation Fees, etc.	38,206.40
Reserve for	
Fluctuation in Canadian Exchange	9.40
TOTAL LIABILITIES	$ 72,738.44
Net Worth (Funds)	422,190.70
Total Liabilities and Net Worth	$494,929.14

At the end of the fiscal year, October 31, 1955, the Balance Sheet shows the Society's Net Worth as $422,190.70 compared to $370,679.89 on October 31, 1954, a net gain of $51,510.81.

Total cash in New York and Cleveland banks $202,804.18 consisting of $34,195.98 in General Operating Fund, $27,656.14 in Reserve Fund, $289.85 in F. Paul Anderson Fund, $70,579.04 in the Research General Fund, $52,368.87 in Research Reserve, $760.57 in Research Endowment, $5,844.99 in Building Maintenance Reserve Fund, $1,024.57 in the Building Fund, and $10,084.17 in the Homer Addams Fund.

Now in the Custodian Account of the Bankers Trust Co., New York, there are securities of the United States and Canadian Governments and a utility which cost as follows: General Fund $3,782.36; Building Fund $37,164.00; Reserve Fund $88,166.50 and F. Paul Anderson Fund $1,000.00.

The Society's Reserve Fund was increased by the addition of $8,712.65 Admission Fees and $1,564.37 Interest earned on Savings Bank Accounts and Securities. This is an increase of $10,277.02 bringing the Reserve Fund to $117,926.16.

The Research Reserve Fund was increased by the addition of $816.39 Interest Earned on Savings Accounts and $63,449.83 Exposition Funds allocated to Reserve Fund. During the year $25,000.00 of this fund was transferred to Research General Fund. As of October 31, 1955 the Research Reserve Fund had a balance of $52,368.87.

The Sources of Society Income were: Admission Fees and Dues $150,684.70; Publications $200,527.78; Interest $908.55; From Research $183,324.36; Exposition Contributions $63,449.83—Total $598,895.22.

Society Expenditures were: Meetings, Committees and Chapters $36,754.05; Publications $152,478.34; Headquarters $170,226.68; Research $203,374.20—Total $562,833.27.

The insurance coverage of the Society is as follows: Fire for Buildings $201,000.00; for Contents $144,500.00; Sprinkler $12,000.00; Automobile Comprehensive Public Liability $100,000-300,000.00; Property Damage $25,000.00; Medical Payments $1,000.00; Bodily Injury $25,000-100,000.00; Fidelity Bonds on Officers and all Employees $20,000.00. Workmen's Compensation is carried in New York and Ohio as required by law.

The Society has contributed to the Employees Retirement Plan Trust $3,360.99 and the participating employees contributed a substantially similar amount deducted from salary.

Membership

The Society admitted the following members in the grades indicated since January 1, 1955:

Life Members	24
Members	328
Affiliates	427
Associates	16
Junior Members	292
Students	127
TOTAL	1,214

The names and addresses of the candidates for membership were published in the JOURNAL of the Society each month during the year and are on record in the Secretary's office. The present membership total in 10,562.

Respectfully submitted,

JOHN E. HAINES, *President*

E. R. QUEER, *Treasurer*

President Haines next introduced Prof. B. H. Jennings, Evanston, Ill., who presented the Report of the Committee on Research for 1955.

ANNUAL REPORT OF COMMITTEE ON RESEARCH—1955

Continued growth and expansion of the research program of the Society took place during the fiscal year, November 1, 1954 to October 31, 1955. Ten projects at the Laboratory in Cleveland, and 15 projects at cooperating university and industrial laboratories were supported during the year. Seventeen technical papers along with the preparation of new and revised data for the HEATING, VENTILATING, AIR CONDITIONING GUIDE resulted from the work. Nine research projects were completed during the year.

Committee activity reached a new high as 5 meetings of the Committee on Research and the Research Executive Committee; 26 meetings of the 23 Technical Advisory Committees, and several subcommittee meetings were held. More than 250 engineers and industrialists thus contributed to the research program, and some of their activities are reported here in detail under appropriate TAC listings.

EXPENDITURES AND FINANCES

Research expenditures totalled $205,000, an increase of 33 percent over 1954. Research income met expenses by virture of a 43-percent increase in the number of industrial contributors, by increased membership dues, and from the larger revenue which accrued

from the biennial exposition sponsored by the Society. The supporting companies, consultants and trade associations who contribute funds and equipment on an annual basis totalled 190 during 1955.

The outlook for 1956 research is optimistic, because of good business conditions and a broadening realization of the benefits of ASHAE research to industry. Several new projects that should attract additional industrial support are under consideration at the year's end, while all of the continuing projects also enjoy a strong interest among the membership and industry.

Participation in the programs is invited by the Committee on Research. With the assistance of the staff and committees, the director of research extends current programs and prepares new programs in cooperation with interested groups. Cost estimates and finances are arranged, and the work is conducted at the Society's Laboratory or at cooperating institutions on a non-profit basis.

AIR CLEANING

The project on air contaminants at the University of Minnesota has been continued by K. T. Whitby under the direction of A. B. Algren. Accomplishments include the completion of a centrifugal sedimentation apparatus and a usable procedure for determining the size ranges and amounts of particulate matter in dust samples. A mixture of subsieve particles from 100 microns down to 0.05 microns can be classified rapidly in one operation in the apparatus. The tool will be useful to researchers and manufacturers of air-cleaning equipment.

Suitable filters for atmospheric dust and lint have been developed and reported. An apparatus and method for studying dusts photometrically has also been evolved and the limitations of such methods have been established.

The project has been pursued vigorously to unravel the problems of dust agglomeration and measurement. Development of accurate methods of evaluating air-cleaning equipment is a prime objective.

A seminar on air cleaning was held at the University in March, and several other meetings were also held in which manufacturers of air filters participated.

AIR DISTRIBUTION

Five projects on air flow and distribution in air conditioning were in progress, 2 of which were completed and 4 publications resulted.

Case Institute of Technology: G. L. Tuve and Alfred Koestel completed a paper on an extensive investigation of the performance of 5 room-air distribution systems having grilles at various levels and locations. The flow patterns in the room and the temperature distributions were plotted to show practical effects. As air delivered to rooms issues in a multiplicity of jets, the results reported for 2 parallel jets are basic to the subject. The jet velocities decrease from the outlet as the jet streams expand and merge. Quantitative data on the flow patterns and velocities are ready for early publication.

During the summer of 1955, tests were conducted on space heating with 95 F air, such as could be produced by heat-pump systems operating at a high coefficient of performance. Larger air volumes were used than in normal warm-air heating, and efforts were directed toward draft-less air flow. Air distribution was by 3 different systems: ceiling diffusers, high sidewall registers, and vertical discharge floor registers.

The same 3 systems were tested for summer cooling. The combined results will be reported soon.

Kansas State College: Linn Helander and associates continued the study of downward projection of heated air as from unit heaters and grilles. One paper reported the jet throw and flow pattern for an annular outlet and for a shallow diffuser. The information is useful to the manufacturers of unit heaters and to the designers of air-distribution systems. The work is expected to be completed in 1957.

Michigan State College: L. G. Miller, C. H. Pesterfield, and R. J. Waalkes completed the analysis of their data on friction losses in rectangular air-duct takeoff elbows and completed the paper on the subject. The friction losses were shown to vary in a complex

manner with the proportion of the air flowing through the takeoff elbow. The project and paper provided valuable design data in a form directly usable by application engineers.

COMBUSTION

Battelle Memorial Institute is in its second year of investigation of the causes and cures for pulsation and resonance noises from domestic heating furnaces and boilers fired by oil and gas burners. It has been shown that the burners, combustion chamber, and flues form an acoustical system with the flame as the driving element. High-speed motion pictures have shown the dancing of the flames in harmony with the noise.

Numerous factors of construction and air-fuel ratio adjustments affect the results. The study which is expected to continue into 1957 will develop design criteria.

COOLING LOAD

The calculation of air-conditioning cooling loads requires consideration of the periodic nature of sunshine and air temperature variations as well as the heat storage and release from structural masses. Two projects were completed in this field at the Research Laboratory and the results were reported and published.

The periodic heat flow through insulated concrete roof sections was measured in the solar calorimeter. The heat flow was also calculated by the Mackey-Wright equations. The results show that the equations are essentially correct and useful, and that the simplifying assumptions used in the equations were satisfactory. The tests were necessary to check the reliability of the equations.

The selection of design conditions from weather data was investigated in 1954 using data for New Orleans. The design conditions selected must be different for office buildings, theaters, and homes, because of the different types of loads and hours of occupancy. A method was demonstrated for tabulating solar radiations, dew-point and dry-bulb temperatures and their frequencies of occurrence. The selection of the design conditions can then be made for various cases. The project was completed and a paper published in April.

HEAT FLOW THROUGH GLASS

Proper skylighting requires an optimum combination of heat exclusion, light transmittance, and light distribution. Skylights contribute cooling load to structures.

Continuing from 1954, the periodic heat gain of structures from skylights was investigated by tests in the solar calorimeter at the Research Laboratory. Both prismatic and diffusing-type skylights were tested. Work has also been done on the performance of slat-type skylight shading devices.

The first publication was for presentation at the January, 1956 Annual Meeting. It provides the essential data for predicting the heat-flow performance for 4 light-controlling skylights and 3 sheet-glass skylights. The instantaneous cooling load can be found for any combination of sun position, skylight orientation, and outdoor air temperature. A second paper is planned to follow shortly as the testing is completed.

HOT WATER AND STEAM HEATING

University of Illinois: W. S. Harris completed the installation of a residential-type hot-water heating system with special facilities for studying gas formation, migration and venting. Preliminary results show the formation of gas by corrosion, by release of dissolved air in the water charged to the system, and by decomposition of dissolved minerals in the water. The migration of air from the compression (expansion) tank by alternate solution in and release from the boiler water was also observed. Oxygen in the gas gradually combines with the iron piping and boiler. The study is continuing.

A second phase of the project is a study of air entrainment in the water flowing through a forced-circulation system having tubular convectors. Glass inspection sections are incorporated in the piping system to permit observation of the gas bubbles.

Northwestern University: Noises generated in pumps and water piping are a source of nuisance when these vibrations become audible, as from pipe and valve chattering or hissing.

Under the direction of W. L. Rogers, a third progress report for publication was written during the year. This covers new work in which a microphone was installed in 1½-in. straight pipe and the intensity of noise produced by the flowing water was found to be a function of the intensity of pressure drop. The frequency of water noise was mainly in the range of 20 to 75 cps (cycles per second), with a negligible part above 300 cps. Cavitation at orifices caused a marked increase in water noise. Water temperature alone has little effect except as it contributes to cavitation. Pump and fitting noises were also explored.

Research on the metastable state of water is being continued by B. H. Jennings. A new metastable smooth-wall test chamber has been constructed in which the metastable state can be established with a minimum of difficulty. Shock patterns which have reached 4 psi have been read on mercury manometers, but it is believed that the inherent pressure impulses are greater than this in magnitude. Work is actively continuing with the accumulation of a large mass of data on various aspects of the phenomenon.

HUMAN CALORIMETRY

The human calorimeter developed at the Research Laboratory and installed at the Naval Medical Research Center at Bethesda, Md., was essentially completed during the year. R. G. Huebscher from the Laboratory was in charge of the construction. Demonstration runs during September 1955 showed that the accuracy and performance met design expectations.

The instrument is unique in many design features. It will measure separately the heat liberation from the skin and respiration of an enclosed subject to an accuracy of 2 to 3 Btu an hour. The unit will be used for studies of metabolism, muscle efficiency and human performance under a wide range of environments.

INFILTRATION

Air infiltration to modern residences and buildings is a significant factor in heating and cooling loads but is understood for precise prediction only in vague terms.

University of Illinois: H. T. Gilkey and D. R. Bahnfleth made a number of infiltration tests during the 1954–55 heating season on a one-story residence with basement. The studies were also continued during summer months. The tests were run by injecting a quantity of helium into the house air and noting the decline in helium concentration as air leaked into and out of the house.

Preliminary results reported for the winter season indicated infiltration of about 0.6 air change per hour for this structure. Wind velocity and direction, and indoor-outdoor temperature difference are among the factors affecting the results. Tests are also to be run on a two-story residence before a final report is prepared.

INSULATION

In cooperation with the TAC on Insulation, D. J. Vild of the Research Laboratory collected and tabulated much new data on heat transfer through building sections. In completion of the assignment, the new computed U values of overall heat transfer have been assembled in a quickly usable form for THE GUIDE, 1956.

ODORS

The effect of air temperature and humidity upon odor perception was investigated at the Research Laboratory using pure compounds and tobacco smoke as odorants. The project was completed with a paper showing significant results.

An increase in humidity generally reduced odor perception. For example, a *definite* odor of pyridine at 20 percent RH was barely *perceptible* at 80 percent RH. Both

observations are the averages of numerous observations by different people while the odorant concentration remained the same in air at 80 F. Humidity did not affect the perception of tobacco odor to the same degree.

Increasing air temperature generally caused a slight decline in the intensity of the odor. Olfactory adaptation to odors and the relation of concentrations to perception were also measured. Future work is to include odor adsorption and release from room furnishings, and a study of methods of odor control.

PANEL HEATING AND COOLING

During 1955, test work was completed on the heat transfer to and from heated and cooled panel surfaces of a room. The work was conducted in a special room 12 × 24.5 ft in the Research Laboratory. The design procedure for heating with ceiling panels using warm water was completed in cooperation with a subcommittee. The paper is for presentation at the 1956 Annual Meeting in Cincinnati. The procedure reported has been simplified as much as deemed practicable, and should be readily usable as well as of permanent value to the profession and the public.

A second paper was prepared relating the factors and coefficients of radiation and natural convection in a panel-heated room.

Tests of heat transfer from fluorescent lighting fixtures to a cooled ceiling panel were also completed and will be reported in the near future. As was known, some of the energy is transferred directly by radiation without first heating the room air, but good data were desired for design use in the cooling of rooms having high lighting intensities.

A program is presently under way to investigate the heat transfer between an outside cold wall of a room and the heated ceiling while the room-air temperature is held constant.

SENSATIONS OF COMFORT AND PHYSIOLOGICAL RESEARCH

During 1955 no projects were sponsored in the field of human response to thermal environment, but plans were made to begin a study of the effect of radiation and of humidity on human comfort. Early in 1956 a program is also to be developed for an investigation of the effect of thermal environment on human performance and productivity. Such research is urgently needed to lay the groundwork for the future development of year-round air conditioning in homes and industry.

SOLAR ENERGY COLLECTORS

University of Minnesota: A program is now in progress under the direction of R. C. Jordan to study experimentally the solar heat transfer to a flat-plate collector, and the transfer from the collector plate to a fluid. The energy is available the year around even in cold climates and could be made available for household purposes by heat pumps and other means.

Two collectors with plates 7 ft-1 in. wide by 3 ft-3 in. high have been installed and are tilted to the south. A 50-50 ethylene glycol-water mixture is the fluid pumped through tubing attached to the plates. Heat transfer and recorder equipment have also been installed. Test results have not yet been received.

SORPTION

Pennsylvania State University: To develop the technology of moisture adsorption by solid sorbents, basic tests were conducted in 1954 on beds of silica gel under dynamic conditions. The research resulted in a paper presented in April, 1955 which aids in predicting the behavior of such adsorbents in practical dehumidifiers. Similar work on other sorbents is being planned by the TAC for a new project. A preliminary test code for laboratory testing of solid-sorption dehumidifiers was tried out and found to be practical within a permissible error of 6 percent.

SOUND AND VIBRATION CONTROL

Late in 1955 the Research Laboratory began a program to develop a practical method for measuring the noise radiated from fans and allied equipment. Four known methods are to be investigated, after which a method is to be formulated that could be adopted as a standard by trade associations for rating equipment. Decibel levels at various frequencies are being measured. W. F. Kerka is conducting the research under the supervisory direction of C. M. Humphreys.

A second project, on noise measurement in duct systems, is to be started soon.

THERMAL CIRCUITS

The Research Laboratory has made further progress in applying the thermal-circuit technique to calculating diurnal heat flow through glass and through structural surfaces. The application of electronic computers has also been investigated and found promising for rapidly solving the complex equations for a variety of conditions. Papers on these subjects have been presented, and others are in preparation.

University of California: Two papers on the electric-analogue studies of periodic heat flow through single wall sections and into a single-room enclosure were published. The calculated results compare very favorably with actual measurements, giving much encouragement to practical benefits. After the method has been developed for building sections of various kinds, and to single rooms, the method will be extended to multi-room and multiple-floor structures.

WEATHER DATA

In cooperation with the Weather Bureau and Air Force, the TAC is working on the development of new tabulations of weather data for use by air-conditioning engineers and architects.

RESEARCH PAPERS AND REPORTS

During the year, the research program resulted in the preparation of a number of papers and reports, most of which were presented at Society meetings during the year and are printed in full in this volume.

FULL REPORT

Complete text of the Report of the Committee on Research appears on pages 154 to 159, Journal Section, Heating, Piping and Air Conditioning, February, 1956.

President Haines then stated that in the absence of Prof. W. L. Rogers, B. H. Jennings would present the single technical paper programmed for this session. Following presentation and discussion, President Haines adjourned the session at 11:45 a.m.

SECOND SESSION, MONDAY, JANUARY 23, 2:00 P.M.

After opening the session in the Ballroom, First Vice Pres. John W. James introduced successively the authors of 3 technical papers which were presented and discussed. The session was adjourned at 4:05 p.m.

THIRD SESSION, MONDAY, JANUARY 23, 2:30 P.M.

Treas. E. R. Queer presided at this session in the Roof Garden and 3 technical papers were presented. Following this, Treasurer Queer adjourned the session at 4:30 p.m.

FOURTH SESSION, TUESDAY, JANUARY 24, 9:30 A.M.

President Haines asked for the report of the Inspectors of Election immediately after opening the session in the Ballroom. G. V. Sutfin, chairman, reported the following results.

REPORT OF INSPECTORS OF ELECTION

Ballot for Officers	Total
President, John W. James, Chicago, Ill.	1916
First Vice President, P. B. Gordon, New York, N. Y.	1914
Second Vice President, E. R. Queer, University Park, Pa.	1910
Treasurer, R. A. Sherman, Columbus, Ohio	1902
Members of Council (Three-Year Term)	
A. B. Algren, Minneapolis, Minn.	1834
John Everetts, Jr., Philadelphia, Pa.	1810
J. N. Livermore, Detroit, Mich.	1817
John H. Ross, Toronto, Canada	1817
Committee on Research (Three-Year Term)	
F. H. Faust, Bloomfield, N. J.	1814
Ford K. Hick, Chicago, Ill.	1814
R. C. Jordan, Minneapolis, Minn.	1809
H. A. Lockhart, Morton Grove, Ill.	1801
Arthur Nutting, Louisville, Ky.	1796
TOTAL BALLOTS RECEIVED	1923
TOTAL LEGAL BALLOTS	1916
INVALID BALLOTS	7

Scattering votes: Members of Council 8 and Committee on Research 1.

Respectfully submitted,

G. V. SUTFIN, *Chairman*
JOHN A. GUY
HERMAN WORSHAM

President Haines turned the session over to Vice Chairman John Everetts, Jr., Philadelphia, Pa., and 3 technical papers were presented and discussed. President Haines adjourned the session at 12:00 noon.

FIFTH SESSION, TUESDAY, JANUARY 24, 10:00 A.M.

After opening the session in the Roof Garden, First Vice President James introduced Prof. M. K. Fahnestock, Urbana, Ill., as vice chairman. Three technical papers were presented and discussed. Vice President James then adjourned the session.

SIXTH SESSION, WEDNESDAY, JANUARY 25, 9:30 A.M.

Meeting in the Ballroom, Second Vice Pres. P. B. Gordon called the sixth session to order and stated that G. V. Sutfin, Cincinnati, Ohio, would report for the Inspectors on the vote for By-Law Amendments.

Mr. Sutfin then gave an item-by-item report of the *yes* and *no* votes on the 28 items on which Members had voted by letter ballot.

REPORT OF INSPECTORS ON VOTE ON BY-LAW AMENDMENTS

Item No. 1	1,737 yes votes, 21 no votes
Item No. 2	1,836 yes votes, 18 no votes
Item No. 3	1,834 yes votes, 18 no votes
Item No. 4	1,834 yes votes, 15 no votes
Item No. 5	1,836 yes votes, 21 no votes
Item No. 6	1,832 yes votes, 19 no votes
Item No. 7	1,840 yes votes, 15 no votes
Item No. 8	1,834 yes votes, 11 no votes
Item No. 9	1,833 yes votes, 20 no votes
Item No. 10	1,843 yes votes, 18 no votes
Item No. 11	1,831 yes votes, 17 no votes
Item No. 12	1,832 yes votes, 15 no votes
Item No. 13	1,834 yes votes, 11 no votes
Item No. 14	1,846 yes votes, 13 no votes
Item No. 15	1,848 yes votes, 12 no votes
Item No. 16	1,851 yes votes, 21 no votes
Item No. 17	1,842 yes votes, 21 no votes
Item No. 18	1,846 yes votes, 12 no votes
Item No. 19	1,851 yes votes, 10 no votes
Item No. 20	1,850 yes votes, 11 no votes
Item No. 21	1,847 yes votes, 16 no votes
Item No. 22	1,842 yes votes, 15 no votes
Item No. 23	1,854 yes votes, 9 no votes
Item No. 24	1,847 yes votes, 10 no votes
Item No. 25	1,846 yes votes, 13 no votes
Item No. 26	1,849 yes votes, 11 no votes
Item No. 27	1,843 yes votes, 13 no votes
Item No. 28	1,841 yes votes, 16 no votes

Respectfully submitted,

G. V. SUTFIN, *Chairman*
JOHN A. GUY
HERMAN WORSHAM

The By-Law changes approved by this result of the voting are:

1. **Article II, Membership:** *Sections 1, 2,* and *3(a)* are amended; a new *Section 3(d)* is added; existing *Section 3(d)* is renumbered as *Section 3(e)*, with first paragraph amended; existing *Section 3(e)* is renumbered as *Section 3(f)* and amended; *Section 3(h)* is amended; *Section 4, 5* and *6* are amended; *Section 8(a)* is amended.
2. **Article III:** Heading amended to read **Chapters, Special and Student Branches and Regional Areas;** existing *Sections 1, 4* and *5* renumbered as *Sections 1, 2* and 3; a new *Section 4* is added; existing *Section 2* is renumbered as *Section 5* existing *Section 3* is renumbered as *Section 6*, and amended; existing *Section 6* is renumbered as *Section 7.*
3. **Article IV, Funds:** *Sections 1, 3* and *5* are amended.
4. **Article V, The Council:** *Section 1* and *Section 2* are amended.
5. **Article VII, Committees:** *Sections 2, 3(a), 3(c),* and *3(f)* are amended; *Section 3(g)* and Rules 1 and 2 are eliminated; *Section 5(c)* is amended.
6. **Article VIII, Meetings, Nominations and Elections:** *Section 3* is amended.
7. **Appendix A** of existing By-Laws is eliminated.

The revisions and amendments are included in the chapter "Charter and By-Laws," #1556, beginning on page 23 of this Volume.

Chairman Gordon introduced the authors of 3 technical papers which were presented and discussed. He then adjourned the session at 11:35 a.m.

SEVENTH SESSION, WEDNESDAY, JANUARY 25, 2:00 P.M.

The seventh session was arranged in the form of a symposium on the general subject of High-Velocity Air Distribution.

Second Vice Pres. P. B. Gordon, New York, N. Y., opened the session in the Ballroom. Five papers had been arranged covering various topics as follows: *Duct Design*—C. M. Wilson, New York, N. Y.; *Economics, Costs* vs. *Velocity*—John Everetts, Jr., Philadelphia, Pa. *Fan and Air Noises*—R. D. Lemmerman, Hartford, Conn.; *Duct Construction*—K. A. J. Monier, San Antonio, Tex.; *Field Testing*—R. D. Tutt, New Britain, Conn.

However, Mr. Lemmerman was not able to be present, and his place was taken by D. B. Callaway, Beverly Hills, Calif. Similarly Mr. Tutt found it impossible to be present and was represented by W. L. Batchelor, New Britain, Conn.

Chairman Gordon called on these authors successively and they presented their prepared material after which the chairman indicated that questions from the floor could be directed to these various authors.

There is no printed record of the floor discussions which took place. Abstracts of the papers presented at this Symposium session appear on pages 145 to 150 of the JOURNAL SECTION, April 1956. On those pages too, a Bulletin is described which contains the papers in full together with all tables and illustrations.

Following completion of the presentation of the Symposium material, Chairman Gordon turned the session back to President Haines.

At this point, President Haines received from F. J. Kurth, New York, N. Y., a plaque presented to ASHAE by a group of German engineers. President Haines accepted the plaque in the name of the Society, expressing appreciation for this thoughtful action.

President Haines called on L. N. Hunter, Johnstown, Pa., who presented resolutions prepared by the Special Resolutions Committee concerning the death of Frank P. Keeney, Chicago, Ill. The resolutions were accepted and made a part of the record.

John Everetts, Jr., Chairman of the Resolutions Committee, presented its reports, which was accepted.

RESOLUTIONS

WHEREAS, the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS is now drawing to its close, and

WHEREAS, this meeting was held in Cincinnati and its Kentucky suburbs and has brought to those who *neffer vere in Zinzinnati* the opportunity to realize their lifelong desires and to those who *before vere in Zinzinnati* an ever repeated delight, and

WHEREAS, the technical life of the Society has been advanced through the presentation and discussion of a series of well prepared papers and symposia, and

WHEREAS, through the excellent facilities provided, the technical and other committees of the Society have been able to conduct their meetings and accomplish their aims efficiently and comfortably, and

WHEREAS, the *Zinzinnati* Chapter, through its Committee on Arrangements, who placed feathers in their hats before the meeting started, have gone far beyond the call of duty in provision of opportunity for good fellowship, high quality of entertainment, and restful relaxation; therefore

BE IT RESOLVED that the sincere thanks and appreciation of all present be extended

TO A. W. Edwards, general chairman, H. E. Russell, vice chairman, H. E. Sproull, honorary chairman, and through them to the entire Committee on Arrangements and to the officers and members of the Cincinnati Chapter,

TO the ladies of the Chapter for their gracious hospitality at the reception and tea and their untiring efforts in entertaining and assisting the visiting ladies,

TO F. W. Giesel of the *Cincinnati Post* for his outstandingly inspirational and thought provoking address at the Welcome Luncheon,

TO the Cincinnati Gas and Electric Co., the Strietmann Biscuit Co., and the Terrace Plaza Hotel for their hospitality on the trips of inspection,

TO the Reverend Laurence Hall, whose talk at the Annual Banquet we look forward to with great anticipation,

TO the hotel managements who have been thoughtful of our needs and comfort,

TO the authors and discussers of technical papers for the educational contributions,

TO President John E. Haines, whose untiring work through the past year has culminated in such a successful Annual Meeting,

TO all members of the Headquarters Staff both present at Cincinnati and those remaining in New York, without whose continuous efforts the progress of the Society would cease, and

TO the chairman of the Committee on Resolutions without whose help these resolutions were prepared, and who has not seen them until this reading.

Respectfully submitted,

The Resolutions Committee

JOHN EVERETTS, JR., *Chairman*, Philadelphia, Pa.
R. A. SHERMAN, Columbus, Ohio
W. J. COLLINS, JR., Oklahoma City, Okla.

President Haines then recognized B. W. Farnes, Portland, Ore., chairman of the Membership Committee, who stated that Dr. Milton Stover Eisenhower, President of The Pennsylvania State University, Charles Franklin Kettering, Dayton, Ohio, and Herbert Clark Hoover, Former President of the United States, New York, had been nominated and endorsed for Honorary Membership in the Society in accordance with Article II, Section 3(a) of the By-Laws. (Biographical data are included at the rear of this volume.)

He presented their names for consideration by the Meeting for election to Honorary Membership. President Haines asked for a vote and the motion was carried.

President Haines recessed the session until the Banquet at 7:00 p.m., announcing that the newly elected officers would be installed at that time, and that at the completion of the Banquet, the 62nd Annual Meeting would be adjourned.

EIGHTH SESSION, WEDNESDAY, JANUARY 25, 2:30 P.M.

The eighth session took the form of a symposium type meeting on the subject of Air Pollution. This session was called to order by Pres. John E. Haines in the Roof Garden. After announcing that the session should adjourn promptly so that all present could have an opportunity to attend the business meeting of the seventh session, President Haines turned the session over to Prof. C. H. Pesterfield, E. Lansing, Mich., who acted as vice chairman and as moderator of the symposium.

Arrangements had been made for the assembly of 6 papers, each by well-known experts in the field of air pollution. The list follows: *Health Aspects*—W. C. Cooper, M.D. and A. C. Stern, both of Cincinnati, Ohio; *Smog*—G. P. Larson, Wayne, Pa.; *Regulations*—H. C. Ballman, Pittsburgh, Pa.; *Cleaning*—J. W. May, Louisville, Ky.; *Practical Approach*—C. W. Gruber, Cincinnati, Ohio; *Enforcement*—J. H. Carter, St. Louis, Mo.

Moderator Pesterfield called on these authors successively and each presented his prepared material. Following this, a discussion took place with questions directed to the several speakers.

Somewhat shortened versions of the papers presented by each of the speakers at the Symposium appear in the March, 1956 of the JOURNAL SECTION on pages 156 to 163.

Following completion of the discussions, the chairman recessed the session, suggesting that all Members present attend the business meeting at end of the seventh session.

PROGRAM—62nd ANNUAL MEETING

Sheraton-Gibson Hotel, Cincinnati, Ohio

January 23-25, 1956

Saturday—January 21

10:00 a.m. Executive Committee, (*Parlor I*)
1:30 p.m. Finance Committee, (*Parlor Q*)
1:30 p.m. Public Relations Committee, (*Parlor N*)
2:30 p.m. Research Executive Committee, (*Parlor O*)
3:00 p.m. Chapter Relations Committee, (*Parlor P*)

Sunday—January 22

9:30 a.m. Council Meeting, (*Parlors N & O*)
10:00 a.m. Committee on Research, (*Parlors P & Q*)
10:00 a.m. REGISTRATION, (*Ballroom Foyer*)
3–5 p.m. Reception and Tea

Monday—January 23

9:00 a.m. REGISTRATION, (*Ballroom Foyer*)
10:00 a.m. FIRST SESSION, (*Ballroom*)
Call to Order—John E. Haines, *President*
Greetings by A. H. Gerdsen, *President*, Cincinnati Chapter
Report of President—John E. Haines
Report of Council—A. V. Hutchinson
Report of Treasurer—E. R. Queer
Report of Committee on Research—B. H. Jennings, *Chairman*
Noise Production and Damping in Water Piping, by W. L. Rogers, Evanston, Ill., presented by B. H. Jennings.
12:15 p.m. Welcome Luncheon, (*Roof*)
Chairman: A. W. Edwards
Speaker: F. W. Giesel, Business Manager, Cincinnati Post
Subject: Forward—A Look at Tomorrow
1:00 p.m. Cinerama Holiday (*Capitol Theater*)

2:00 p.m. SECOND SESSION, (*Ballroom*)
Call to Order—John W. James, *Vice-President*
Performance of Covered Hot Water Floor Panels, Part I—Thermal Characteristics, by E. L. Sartain, Racine, Wis., and W. S. Harris, Champaign, Ill., presented by Mr. Sartain.
Thermal Design of Warm Water Ceiling Panels—A Design Manual prepared by a subcommittee of TAC on Panel Heating and Cooling and members of the staff of the ASHAE Research Laboratory, presented by L. F. Schrutrum, Cleveland, Ohio.
Heat Gain Through Glass Skylight Fenestrations, by D. J. Vild and G. V. Parmelee, Cleveland, Ohio, presented by Mr. Vild.

2:30 p.m. THIRD SESSION, (*Roof*)
Call to Order—E. R. Queer, *Treasurer*
Moisture in Transient Heat Flow, by K. R. Solvason, Ottawa, Ont., Canada, presented by Mr. Solvason.
Characteristics of Downward Jets from a Vertical Discharge Unit Heater, by S. M. Yen, Linn Helander and L. B. Knee, Manhattan, Kan., presented by Dr. Yen.
Resistance of Rectangular Divided-Flow Fittings, by L. G. Miller, C. H. Pesterfield, E. Lansing, Mich., and R. J. Waalkes, Holland, Mich., presented by Mr. Waalkes.

6:45 p.m. Kentucky Party at Beverly Hills Country Club

Tuesday—January 24

9:00 a.m. REGISTRATION, (*Ballroom Foyer*)

9:30 a.m. FOURTH SESSION, (*Ballroom*)
Call to Order—John E. Haines, *President*
Report of Inspectors of Election—G. V. Sutfin
John Everetts, Jr., *Vice Chairman*
Evaluation of Equipment Noise, by H. C. Hardy and D. E. Bishop, Chicago, Ill., presented by Dr. Hardy.
Electric Analogue Studies of Single Walls, by Harry Buchberg, Los Angeles, Calif., presented by Professor Buchberg.
Electric Analogger Analysis of a Cooled Structure Complex, by C. F. Kayan, New York, N. Y., presented by Professor Kayan.

10:00 a.m. FIFTH SESSION, (*Roof*)
Call to Order—John W. James, *Vice-President*
M. K. Fahnestock, *Vice Chairman*
Index for Evaluating Heat Stress in Terms of Resulting Physiological Strains, by H. S. Belding and T. F. Hatch, Pittsburgh, Penn., presented by Dr. Belding.
Body Evaporation During Short Exposures to Various Temperatures, Humidities, Pressures and Mass Velocities, by J. W. McCutchan, Los Angeles, Calif., presented by Mr. McCutchan.
Humidity Effects on the Odor Problem, by R. L. Kuehner, York, Penna., presented by Dr. Kuehner.

10:30 a.m. Ladies City Trip and Luncheon at Hotel Alms

12:15 p.m. Life Members' Dutch Treat Luncheon, (*Room 7*)

1:30 p.m. Chapters Conference Committee, (*Roof*)

1:30 p.m. TAC on Air Cleaning, A. B. Algren, *Chairman*, (*Parlors P & Q*)

1:30 p.m. TAC on Air Distribution, W. O. Huebner, *Chairman*, (*Parlor I*)

1:30 p.m. TAC on Physiological Research, M. K. Fahnestock, *Chairman*, (*Parlor N*)

1:30 p.m. TAC on Sorption, G. L. Simpson, *Chairman*, (*Parlor O*)

2:00 p.m. Inspection Trips: (1) Beckjord Station; (2) Strietmann Biscuit Co.; (3) Terrace Plaza Hotel

7:00 p.m. Past Presidents' Dinner, (*Parlor I*)

7:30 p.m. TAC on Combustion, H. R. Limbacher, *Chairman*, (*Parlor N*)

7:30 p.m. TAC on Industrial Environment, P. J. Marschall, *Chairman*, (*Parlor E*)

7:30 p.m. TAC on Odors, T. H. Urdahl, *Chairman* (*Parlor O*)

7:30 p.m. TAC on Thermal Circuits, S. F. Gilman, *Chairman*, (*Parlor P*)

7:30 p.m. TAC on Weather Data, John Everetts, Jr., *Chairman*, (*Parlor Q*)

8:00 p.m. Chapter Publicity Chairmen—Editors Meeting, J. H. Fox, *Chairman*

Wednesday—January 25

9:00 a.m. REGISTRATION, (*Ballroom Foyer*)

9:30 a.m. SIXTH SESSION, (*Ballroom*)
Call to Order—P. B. Gordon, *Vice-President*
Report of Inspectors on Vote on By-Law Amendments—G. V. Sutfin
Elements of Dual-Duct Design and Performance, by N. S. Shataloff, New York, N. Y., presented by R. W. Waterfill.
Branch Fitting Performance at High Velocity, by C. M. Ashley, S. F. Gilman and R. A. Church, Syracuse, N. Y., presented by Dr. Gilman.
Self-Actuated Room Control from High Speed Air, by E. F. Snyder, Jr., Minneapolis, Minn., presented by Mr. Snyder.

12:30 p.m. Ladies Style Show and Luncheon, (*Cincinnati Club*)

2:00 p.m. SEVENTH SESSION, (*Ballroom*)
Call to Order—P. B. Gordon, *Vice-President*
HIGH VELOCITY AIR DISTRIBUTION SYMPOSIUM
Duct Design—C. M. Wilson
Economics, Costs *vs.* Velocity—John Everetts, Jr.
Fan and Air Noises—R. D. Lemmerman
Duct Construction—K. A. J. Monier
Field Testing—R. D. Tutt
Pres. John E. Haines, presiding
Report of Committee on Resolutions
Unfinished Business
New Business
Recess

2:00 p.m. EIGHTH SESSION, (*Roof*)
Call to Order—John E. Haines, *President*
AIR POLLUTION SYMPOSIUM
C. H. Pesterfield, *Vice Chairman*
Health Aspects—W. C. Cooper, M.D.
Smog—G. P. Larsen
Regulations—H. C. Ballman
Cleaning—J. W. May
Practical Approach—C. W. Gruber
Enforcement—J. H. Carter
Recess

4:00 p.m. Nominating Committee, (*Parlor E*)

6:30 p.m. Social Hour, (*Roof*)

7:00 p.m. ANNUAL BANQUET, (*Roof*)
Toastmaster: C. P. Krantz
Introduction and Installation of 1956 Officers
Presentation of Past President's Award to John E. Haines

Presentation of F. Paul Anderson Medal to R. W. Keeton, M.D.
Speaker: The Reverend Laurence Hall, British-American Humorist
Subject: Humor—America's Secret Weapon
Adjournment
Music for Dancing

Thursday—January 26

9:30 a.m. Council Meeting, (*Parlor E*)
(organization meeting of 1956 Council)

10:00 a.m. Committee on Research, (*Room 4*)
(organization meeting of 1956 Committee)

10:00 a.m. Guide Committee, (*Parlor G*)

COMMITTEE ON ARRANGEMENTS

A. W. Edwards, *General Chairman*
H. E. Russell, *Vice Chairman*
H. E. Sproull, *Honorary Chairman*

Banquet Committee—E. A. Sobolewski, *Chairman*, R. A. Albrecht, Jr., R. H. Endejann, J. F. Leisgang, M. E. Mathewson, R. S. Rose, D. E. Tullis.

Entertainment Committee—C. P. Krantz, *Chairman*, R. B. Barbara, D. L. Bergman, B. R. Byard, C. J. Eubank, W. A. Fennell, Ludwig Jensen, E. J. Kramer, J. A. Lamb, W. B. Motz, Pinkney Varble.

Finance Committee—E. W. McNamee, *Chairman*, J. J. Bechtol, A. H. Gerdsen, I. B. Helburn, Wm. H. Junker, T. D. Reiley, G. V. Sutfin.

Ladies Committee—R. C. Beineke, *Chairman*, H. A. Pillen, *Vice Chairman*, R. B. Walcott. Assisted by: Mmes. R. C. Beineke, A. W. Edwards, A. H. Gerdsen, Isidore Jacobson, C. P. Krantz, E. W. McNamee, H. A. Pillen, T. D. Reiley, C. A. Russell, E. A. Sobolewski, G. V. Sutfin, F. W. Wilson.

Publicity Committee—R. G. Anderson, *Chairman*, T. J. Furlong, *Vice Chairman*, L. R. Craig, F. B. De Bra, G. B. Houliston, L. G. Powers, E. J. Richard, C. D. Weaver, Jr.

Reception Committee—F. W. Wilson, *Chairman*, J. J. Bechtol, Vincent Hotze, H. K. Jennings, K. B. Little, J. M. Schweiger, A. J. Staubitz, P. P. Zillick.

Sessions Committee—H. W. Moore, *Chairman*, F. J. Bottenhorn, III, E. A. Carsey, J. A. Diehl, W. H. Fogarty, R. A. Frey, F. S. Hamer, W. A. Juergens, F. H. Kiesewetter, F. B. Lienhart, N. O. Mitchell, R. L. Moore, O. W. Motz, W. H. Rieger, Joe Stockwell, Herman Worsham.

Transportation Committee—N. A. Frankel, Jr., *Chairman*, L. P. Brehm, Jr., J. W. Gibbs, Isidore Jacobson, R. F. McAndrews, Judson Prindle.

1556

No. 1556

CHARTER

Provisions of

CERTIFICATE OF INCORPORATION

of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

(As filed on January 24, 1895 and amended by Certificates of Amendment filed on or about May 20, 1914, March 8, 1946, December 7, 1949, February 1, 1952 and December 8, 1954.)

1. The corporate name of the Society is to be known as AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

2. The certificate of incorporation was filed in the office of the Secretary of State on or about January 24, 1895. Amended certificates of incorporation were filed in the office of the Secretary of State on or about May 20, 1914, March 8, 1946, December 7, 1949, February 1, 1952 and December 8, 1954.

3. The number of directors shall be not less than thirteen (13) nor more than twenty-one (21). The Board of Directors shall be styled "The Council."

4. The purposes for which this corporation is organized are:

(*a*) To advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

(*b*) To encourage and conduct scientific research and the study of principles and methods in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, the results of which shall be made freely available to the public.

(*c*) To promote the unrestricted dissemination of knowledge and information and, for such purpose, to publish and to foster the publication of books, periodicals, reports, educational programs and scientific and technical data relating to heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*d*) To engage in educational activities (not including the conduct of any school, schools or institutions of learning), and to encourage the adoption and maintenance of high standards of instruction and technical and professional training in the fields of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

(*e*) To cooperate with governmental agencies and with universities, colleges, schools and other organizations and groups having the same or similar objects and purposes, and to establish scholarships and make contributions, grants and awards in furtherance of the foregoing purposes.

(*f*) To assist in the formation of local chapters and student and other branches, and to regulate, operate and control the same under the direction and at the pleasure of the corporation, but no local chapter or branch shall subject the corporation to any financial or other obligation except such as the corporation may voluntarily assume.

(*g*) To receive, acquire, hold and maintain any property, real or personal, without limitation as to amount or value, for any of the corporation's objects, by way of bequest, devise, gift, purchase or lease, to invest and reinvest the same, to control the income therefrom, and to expend or otherwise dispose of all or any portion of its funds and property, including the income, interest or principal.

(*h*) To do any and all things necessary or proper in connection with or incidental to any of the foregoing.

(*i*) This corporation shall be operated exclusively for scientific and educational purposes; no substantial part of the activities of this corporation shall be the carrying on of propaganda or otherwise influencing or intending to influence legislation; in the event of the dissolution of the corporation, the Council (directors) shall dispose of its net assets, in trust, however, to further the purposes expressed herein, without preference in favor of any contributor or any member, officer or director of the corporation.

5. The principal place of business or office of said Society is to be in the City and County of New York.

(*a*) The territory in which the corporation's operations are principally to be conducted is in all parts of the United States and its possessions and in the Dominion of Canada, but the corporation may on occasions extend its operations to other parts of the world.

(*b*) No officer, director or member of this corporation shall receive or be lawfully entitled to receive any part of the net earnings thereof or any pecuniary profit from the operations thereof, except such reasonable compensation for services in effecting one or more of its purposes as the Board of Directors may determine.

(*c*) Each director of the corporation shall be indemnified by the corporation against expenses actually and necessarily incurred by him in connection with the defense of any action, suit or proceeding in which he is made a party by reason of his being or having been a director of the corporation, except in relation to matters as to which he shall be adjudged in such action, suit or proceeding to be liable for willful negligence, misfeasance or misconduct in the performance of his duties as director; such right of indemnification shall not be deemed exclusive of any other right to which he may be entitled under any by-law, agreement, vote or otherwise.

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BY-LAWS

of

AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC.

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ARTICLE I

Organization

Section 1. ***Organization.*** AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC. is a membership corporation, organized and existing under and by virtue of the laws of the State of New York.

Section 2. ***Object.*** The object of the Society is to advance the arts and sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, for the benefit of the general public.

Section 3. ***Members.*** The rights and privileges of a member of the Society shall be personal to himself and shall not be delegated or transferred, except that each member entitled to vote may vote in person or by proxy as hereinafter provided.

Section 4. ***Government.*** The Society shall be governed by its Charter and these By-Laws, the Rules promulgated by the Council in harmony therewith, and any amendments to the foregoing.

ARTICLE II

Memberships

Section 1. ***Grades.*** The grades of membership in the Society shall be designated as follows: (*a*) Honorary Members, (*b*) Presidential Members, (*c*) Life Members, (*d*) Fellows, (*e*) Members, (*f*) Associate Members, (*g*) Affiliates, and (*h*) Students.

Section 2. ***Construction.*** Whenever in these By-Laws the words "MEMBER" or "MEMBERS" appear in upper-case letters (*sic*, "MEMBER," "MEMBERS"), they shall be taken to be synonymous with the terms "Honorary Members," "Presidential Members," "Life Members," "Fellows," and "Members"; and whenever the words "member" or "members" appear in lower-case letters (*sic*, "member," "members") and the context permits it, they shall be taken to be synonymous with all the grades of membership in the Society.

Section 3. ***Qualifications.***

(*a*) *Honorary Members.* Any notable person of preeminent professional distinction may be elected an Honorary Member at an Annual Meeting of the Society.

(*b*) *Presidential Members.* Upon the installation of his successor, the outgoing President of the Society shall become a Presidential Member.

(*c*) *Life Members.* A Member, an Associate Member, or an Affiliate, in good standing, who has attained the age of sixty-five (65) years and has paid dues in said grades for thirty (30) years shall become a Life Member.

(*d*) *Fellows.* The grade of Fellow shall have an honorary status, which may be conferred upon a MEMBER of professional distinction but for which he may not apply. A MEMBER who has attained the age of forty-five (45) years, who has been in good standing as a MEMBER for a period of at least ten (10) years prior to the date of his proposal for Fellow grade, and who has attained unusual distinction in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, or in the teaching of a major course in said arts and sciences, or who by reason of invention, research, original work, or as an engineering executive on projects of unusual or important scope, has made substantial contributions to said arts and sciences, shall be eligible to become a Fellow. The grade of Fellow shall not be conferred upon more than ten (10) MEMBERS in any calendar year. The grade of Fellow shall not be conferred upon a member of Council during his term of office.

(*e*) *Members.* Any engineer or teacher of engineering, having ten (10) years or more of experience of a character satisfactory to the Council in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, of which four (4) years or more shall have been in responsible charge of important engineering work or in the teaching of courses in said arts and sciences, shall be eligible to become a Member. Architects, chemists, physicians, scientists, and other persons, deemed by the Council to be qualified by reason of special experience in the said arts and sciences, shall also be eligible to become Members.

Graduation from an engineering school with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from a foreign engineering school maintaining similar standards, or from an engineering school of collegiate grade approved by the Council, shall be deemed equivalent to four (4) years of such experience; from other technical schools maintaining four (4) year courses, three (3) years; from a non-engineering college, two (2) years; and each successfully completed year in an engineering college shall be deemed equivalent to one (1) year of such experience.

(*f*) *Associate Members.* Any person having graduated from a college or school of engineering with an engineering curriculum accredited by the Engineers' Council for Professional Development, or from a foreign engineering school maintaining similar standards, or from an engineering school of collegiate grade approved by the Council, or possessing eight (8) years of experience satisfactory to the Council in the arts relating to the sciences of heating, ventilating, cooling or air conditioning, or the allied arts and sciences, shall be eligible to become an Associate Member. Each successfully completed year in such college or school shall be deemed equivalent to one (1) year of such experience. Upon the adjournment of the meeting* at which this by-law shall be adopted, all Junior Members shall be designated Associate Members.

(*g*) *Affiliates.* Any person over twenty-five (25) years of age, not necessarily an engineer but deemed by the Council to be qualified to cooperate with heating, ventilating, cooling or air conditioning engineers in the advancement of professional knowledge, shall be eligible to become an Affiliate.

(*h*) *Students.* Any person regularly enrolled as a graduate student or as an undergraduate student pursuing an engineering curriculum accredited by the Engineers'

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* January 23–25, 1956.

Council for Professional Development in a college or school of engineering, or in a foreign college or school of engineering maintaining similar standards, or in an engineering school of collegiate grade approved by the Council, shall be eligible for Student membership.

Section 4. ***Privileges and Limitations.*** Honorary Members, Presidential Members, Life Members, and Fellows shall be entitled to all the rights and privileges of a MEMBER. Associate Members thirty (30) years of age or over shall be eligible to vote. Associate Members shall not hold elective office in the Society. Affiliates and Students shall not be eligible to vote or hold elective office in the Society. A student membership shall cease upon the student's graduation, or at the close of the school term during which his enrollment as a student has terminated. The Council may transfer a Student to Associate Member grade on proof of his qualifications therefor, or to Affiliate grade.

Section 5. ***Proposals and Applications.*** Proposals for Honorary membership or for Fellow grade shall be signed by ten (10) MEMBERS and by two-thirds of the members of the Council, and shall set forth resumés of the nominees' qualifications and attainments. Applications for membership in the Society or for advancement to higher grades shall be on written forms approved by the Council, setting forth resumés of the candidates' qualifications, experience and references, and such other pertinent data as the Council and the Committee on Admission and Advancement may require.

Section 6. ***Prerequisites.*** The affirmative vote of fifteen (15) members of the Council shall be required for the election of an Honorary Member, or the conferring of the Fellow grade. A two-thirds affirmative vote of the Council shall be required for the admission of a candidate to membership, or for advancement to a higher grade. No person, except a student, shall be admitted to membership or advanced to a higher grade unless his name shall have been published in an issue of the JOURNAL, or mailed to all members.

Section 7. ***Resignation.*** Any member may resign his membership by writing to the Executive Secretary, who shall present the resignation to the Council at its next meeting. If the dues of the resigning member shall have been paid to the end of the quarter-annual period immediately preceding the date of the receipt of resignation, his resignation shall be accepted.

Section 8. ***Suspension and Expulsion.***

(*a*) *Non-Payment of Dues:* If any Fellow, Member, Associate Member, or Affiliate shall fail to pay his dues by April 1, he shall be classed as delinquent and not entitled to vote; if such dues are not paid by July 1, he shall be classed as not in good standing and his membership shall be suspended; if such dues are not paid by December 1, the Executive Secretary shall notify the suspended member by registered mail that unless such dues are paid by December 31, he shall cease to be a member of the Society, and upon his failure to cure such default by December 31, his membership in the Society shall cease; if any Student shall fail to pay his dues by December 31, the delinquent Student's membership shall cease and the Executive Secretary shall notify such Student by registered mail that his membership in the Society has ceased; provided that upon written application satisfactorily explaining a default, accompanied by payment of dues, the Council may, in its discretion, rescind any forfeiture of membership.

(*b*) *Misconduct:* The Council may, by a two-thirds vote of all the members thereof, censure, suspend or expel any member for misconduct in his relations to the Society, after written preferment of charges, thirty (30) days' written notice of hearing sent by registered mail, and an adequate opportunity to be heard before the Council or a committee of one or more MEMBERS designated by the Council.

Section 9. ***Effect of Termination of Membership and Lapse of Time.*** (*a*) All membership rights and interest in the property of the Society of persons resigning or otherwise ceasing to be members, or during the period of suspension of a suspended member, or on death, shall vest in the Society. (*b*) All acts of the Council which shall have received the express or implied sanction of the Society shall be deemed to be the acts of the Society and shall not thereafter be impeached by any member. (*c*) No claim, de-

mand, action or proceeding (individual, representative or derivative) shall be asserted or instituted by a member or former member against the Society, or any incumbent or former officer, Council or committee member thereof, by reason of service in any such capacities, after the expiration of six (6) months from the time when the same has accrued or actual or constructive knowledge has been acquired, or become available, of the facts upon which the same depends.

ARTICLE III

Chapters, Special and Student Branches, and Regional Areas

Section 1. ***Chapters.*** The Council may establish Chapters, which shall operate under the provisions of the Charter and the By-Laws of the Society, and the Rules and Regulations of the Council.

Section 2. ***Special Branches.*** Special Branches of the Society may be established, operated and maintained under the direction and in the discretion of the Council.

Section 3. ***Student Branches.*** Student Branches of the Society may be established, operated and maintained under the direction and in the discretion of the Council.

Section 4. ***Regional Areas.*** The Chapters and Branches of the Society in North America shall be divided by the Council into seven (7) geographical groups, and each group shall be designated a Regional Area. Delineation of Regional Areas, and changes therein, shall be published in an issue of the JOURNAL or mailed to all members.

Section 5. ***Chapter Membership.*** Chapters shall be composed of at least twenty (20) members of any and all grades, exclusive of Chapter Student members residing in the vicinity of the Chapter's headquarters, and only members of the Society in good standing shall be eligible to become and remain Chapter members. Chapter members shall hold the same grade of membership in the Chapter as are held by them in the Society. No member shall vote or hold office concurrently in more than one (1) Chapter of the Society. All grades of Chapter members, except Students, shall be eligible to vote and hold office in Chapters.

Section 6. ***Limitations.*** The elected officers of Chapters shall receive no salary, emolument or compensation for their services as such. Chapters shall not act for the Society or subject the Society to any financial or other obligation, except such as the Society or the Council may by resolution assume. Notice to the foregoing effect shall be imprinted on the stationery used by each of the Chapters. Each Chapter shall promptly file a copy of its minutes with the Executive Secretary of the Society and make report to said Secretary of all of its proceedings. Chapters and Branches shall file with the chairmen of their respective Chapters Regional Committees their recommendations concerning the policies, procedures and operation of the Society, its Chapters and Branches. No contributions, except dues, shall be received or solicited by Chapters without the written approval of the Council. Chapters shall not issue publications or use the Society's name or emblem or Chapter insignia, without the approval of the Council. Chapters shall give no recommendations, endorsements or approvals of any scientific, literary, mechanical or engineering product for the promotion of private interests.

Section 7. ***Revocation of Charters.*** The charters of Chapters, Special Branches and Student Branches may be revoked by a two-thirds vote of all the members of the Council after written preferment of charges, sixty (60) days written notice of hearing sent by registered mail to the President of the Chapter or Branch, and an adequate opportunity to be heard before the Council or a committee of three (3) or more MEMBERS designated by the Council.

ARTICLE IV

Funds

Section 1. ***Admission Fees.*** Honorary Members and Students shall not be required to pay admission fees. The admission fees of Members, Associate Members and Affiliates shall be fixed by the Council, and shall be published in the JOURNAL. Admission fees shall accompany all applications for membership, and shall be refunded only in the event of the rejection of an application.

Section 2. ***Society Reserve Fund.*** Admission fees and such other funds as may from time to time be recommended by the Finance Committee and allocated by the Council shall be set aside and the principal thereof maintained as a Society Reserve Fund. Unless increased by the Society at an Annual or Special Meeting, the Society Reserve Fund shall be limited to a sum equal to fifteen dollars ($15.00) per member at the close of each fiscal year. The Council is hereby authorized and empowered, in any fiscal year in which the Society's revenues may be insufficient to meet expenses, to utilize a maximum of twenty percent (20%) of the Society Reserve Fund as valued on the first day of the fiscal year in which such a withdrawal may be required.

Section 3. ***Dues.*** Honorary Members, Presidential Members and Life Members shall be exempt from the payment of dues. The conferring of Fellow grade upon such members shall not affect such exemption. Unless changed by the Society at an Annual or Special Meeting, the annual dues of Fellows, Members, Associate Members thirty (30) years of age or over, and Affiliates shall be twenty-five dollars ($25.00). The annual dues of Associate Members under thirty (30) years of age shall be fifteen dollars ($15.00). The annual dues of Students shall be fixed by the Council and shall be published in the JOURNAL.

Annual dues, except Students, shall become due and payable in United States currency, or its equivalent, in advance on January 1 of each year, and shall in no case be subject to refund. Dues of new and advanced members, except Students, shall be prorated on a quarter-annual basis, and shall be payable on the first day of the month following notification of election or advancement, and if not paid within three (3) calendar months after such notification, the election or advancement shall be automatically rescinded. Students' dues shall be applicable to the annual period commencing October 1 and terminating September 30. The Council in its discretion may suspend payment of dues by members in the Armed Forces in time of war, or may in its discretion defer or remit the dues of any member for special cause. Future annual dues may be compounded at a three percent (3%) rate by payment to the Society of the worth of an annuity equal to the member's dues for the period for which dues would be required. Compounded payments shall in no event be subject to refund.

Section 4. ***Publications.*** Members of all grades, except Student and dues paying members whose pro-rated dues amount to less than ten dollars ($10.00), shall be entitled to receive the Society's TRANSACTIONS and GUIDE.

Section 5. ***Allocation of Dues for Research.*** Unless changed by the Society at an Annual or Special Meeting, forty percent (40%) of the dues received from Fellows, Members, Associate Members thirty (30) years of age or over, and Affiliates shall be allocated for basic or fundamental research in the principles and laws underlying matters in the arts relating to the sciences of heating, ventilating, cooling and air conditioning, and the allied arts and sciences.

Section 6. ***Investment of Funds.*** (*a*) The Society Reserve Fund and such other funds as may be allocated by the Council for investment shall be invested and reinvested in bonds and obligations of the United States Government, the Government of the Dominion of Canada, or in investments legal for trust funds under the laws of the State of New York, subject to the proviso that not less than one-half of such Fund

and funds shall be invested in bonds and obligations of the United States Government; (*b*) gifts and bequests to the Society for a specific purpose or purposes shall, after acceptance by the Council, be used for the purposes specified and invested in the manner directed by the donor or testator, or in the absence of such direction, in the manner provided in subdivision (*a*) hereof; (*c*) the Council is authorized and empowered, in its discretion and without liability to the Society or to any member thereof, to retain any gift or bequest in property (real, personal or mixed) in the manner and form in which it shall be at the time of such gift or bequest.

Section 7. ***Budget.*** The Finance Committee shall submit to the Council at the Council's last quarter-annual meeting a Budget of estimated income and expenditures of the Society and all the committees thereof, for the succeeding fiscal year commencing November 1 and ending October 31. The expenditures of the Society's funds shall be governed by the Budget as approved, modified, or from time to time amended by the Council, and no additional expenditures shall be made without the approval of the Council.

Section 8. ***Receipts and Disbursements.*** The Executive Secretary shall render all bills and collect all moneys due the Society, and shall enter all receipts in the Society's books and deposit the same to the Treasurer's Account in banks designated by the Council. Except as hereinafter provided, no contract or other obligation for the payment of money shall be valid unless signed or countersigned by the Executive Secretary. Except as hereinafter provided, all vouchers against the Society for the payment of funds shall be submitted to the Executive Secretary, who shall certify the correctness thereof and the authorization thereof by the Budget, and payment shall be made only upon such certification.

Section 9. ***Bonds.*** The Treasurer, the Executive Secretary, and all other Officers, Agents or Employees authorized by the Council to endorse or execute drafts for the payment of money, shall give bond in a penal sum and with sureties approved by the Council, for the faithful performance of their duties, the premiums therefor, if any, to be paid by the Society.

Section 10. ***Audits.*** Between the close of the fiscal year and January 1 of each year, the accounts of the Society shall be audited by a certified public accountant approved by the Council, and the auditor's report shall be presented by the Treasurer at the Annual Meeting of the Society, and shall be published in the JOURNAL.

ARTICLE V

The Council

Section 1. ***Members.*** The Council shall consist of the last living Past President, the President, the First Vice President, the Second Vice President, the Treasurer and twelve (12) MEMBERS, seven (7) of whom shall be from different Regional Areas and elected as Regional Directors for their respective Areas. The twelve (12) MEMBERS shall be divided into three (3) classes of four (4) in each class, and the MEMBERS in each class shall hold office for three (3) years and until their successors shall have been elected and installed. Four (4) MEMBERS shall be elected to the Council at each Annual Meeting, and also such additional number, if any, as may be necessary to fill vacancies. Pending the Annual Meeting, vacancies occurring in the Council may be filled by the Council. The Executive Secretary of the Society shall be the Secretary of the Council. He may take part in the deliberations of the Council but shall not be a member thereof or have any vote therein.

Section 2. ***Powers.*** In addition to the powers specifically conferred upon the Council (directors) by the laws of the State of New York, the Charter and these By-Laws, the

Council shall have the general management and full control of the affairs and all activities of the Society and, subject to the Charter, the By-Laws and the laws of the State of New York, may, in its discretion, promulgate and amend Rules and Regulations, and issue directives for the administration of the Society's affairs and the regulation of all Committees, Chapters, Branches, Regional Areas, Officers, Agents and Employees. The Council may, in its discretion, refer to the Society any important question pertaining to the Society, and shall refer any such question to the Society upon a majority vote taken at a stated or Special Meeting held by the Society.

Section 3. ***Meeting, Quorum and Reports.*** The Council shall hold its annual meeting as soon as practicable after the close of the Annual Meeting of the Society, and shall hold meetings quarter-annually thereafter. Special meetings may be called by the President or by three (3) Council members. A majority of the Council members in office shall constitute a quorum. The Council shall keep a record of its proceedings, and shall report on its activities at each meeting of the Society and, at the Annual Meeting, it shall present a written report as required by the Membership Corporations Law of the State of New York.

Section 4. ***Notice of Council Meetings.*** Unless waived in writing or by telegraph or cable, notice of any quarterly or special meeting of the Council shall be given in writing, mailed to the last known address of each Council member, by the Executive Secretary or the President, or the three (3) Council members calling the meeting, not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting.

ARTICLE VI

Officers

Section 1. ***Elected Officers.*** The elected officers of the Society shall be a President, a First Vice President, a Second Vice President and a Treasurer. The elected officers shall receive no salary, emolument or compensation for services rendered to the Society, and shall serve for one (1) year and until their respective successors shall be elected and installed.

Section 2. ***Appointed Officers.*** The Executive Secretary shall be appointed by the Council at its annual meeting, at a salary fixed by the Council, to serve for one (1) year and until his successor shall be appointed. The Executive Secretary shall be subject to removal by a two-thirds vote of the Council cast by secret ballot.

Section 3. ***Presiding Officer.*** At all meetings of the Society and of the Council, the President, or in his absence, the Vice Presidents in order of seniority, or in their absence the Treasurer or a MEMBER selected by the Council, shall preside.

Section 4. ***The President.*** The President shall exercise the powers and duties assigned to him by these By-Laws and, subject to the direction of the Council, he shall be the chief executive officer of the Society and generally supervise its affairs. At the Annual Meeting of the Society he shall make a report relative to the Society's condition, activities and progress. No President may be re-elected to that office until the Annual Meeting next following the expiration of his term.

Section 5. ***Vice Presidents.*** In the absence, disability, resignation or death of the President, the Vice Presidents in order of seniority, shall exercise the powers and perform the duties of the President.

Section 6. ***The Treasurer.*** The Treasurer shall have custody of the funds of the Society and the Society's books of account, which shall be open to the inspection of any member of the Council. Pending the Annual Meeting, a vacancy occurring in the office of Treasurer shall be filled by the Council.

Section 7. ***The Executive Secretary.*** The Executive Secretary shall be the chief administrative officer of the Society and the manager of the Society's publications. Copies of the minutes of all meetings of the Society, of the Council, and of all committees shall be filed with him. He shall keep such books, papers and records as the Society or the Council may direct, which shall be open to the inspection of any member of the Society. He shall be the keeper of the seal of the Society, and may in his discretion use the designation "Secretary" on legal documents. He shall conduct the correspondence of the Society and shall keep full records of the same. He shall promptly notify the members of the Council, the officers, the nominated candidates, the members of all committees, and applicants for admission or advancement, of their election, nomination, appointment or advancement. He shall issue notices of all meetings of the Society, and, in the case of Special Meetings, he shall add a brief note of the object of the call. He shall be in charge of the offices of the Society and shall administer them under such rules of procedure as the Council may approve. Subject to the discretion of the Council, the Executive Secretary may employ such assistant secretaries and other personnel as may be deemed to be necessary.

Section 8. ***Other Duties.*** All officers of the Society shall perform the duties customarily attaching to their respective offices under the laws of the State of New York, and such other duties and services incident to their respective offices as are delegated to them in these By-Laws and as may from time to time be assigned to them by the Council.

ARTICLE VII

Committees

Section 1. ***Advisory Board.*** The Advisory Board shall consist of all Presidential Members, of which the last living Past President shall be chairman, and the said committee shall consider and make recommendations to the Council concerning matters of policy affecting the Society referred to the Board by the Council.

Section 2. ***Council Committees.*** Unless otherwise provided, the Council Committees and the respective chairmen thereof shall be appointed by the President, with the approval of the Council, as soon as practicable after the close of the Annual Meeting. Each Council Committee shall consist of three (3) Council members, who shall serve for a term of one (1) year and until their successors are chosen. The following shall be the Council Committees and their respective duties:

(*a*) *Executive Committee,* which shall investigate and make reports and recommendations to the Council regarding matters relating to the Society or any member or members thereof. During intervals between Council meetings the Executive Committee may exercise such powers of the Council as may lawfully be delegated to it by the Council.

(*b*) *Finance Committee,* which under the direction of the Council shall supervise and control the financial affairs of the Society and its books of account. It shall survey, investigate and analyze all financial requirements and expenditures, scrutinize all Budget estimates, and prepare the Budget for submission to the Council. The Treasurer shall be an ex-officio member of the said committee, with the power to vote.

(*c*) *Membership Committee,* which shall publicize the aims, activities, achievements and the scientific and educational purposes of the Society, toward the end that persons duly qualified shall apply for membership therein.

(*d*) *Program and Papers Committee,* which shall plan the general character of all technical meetings of the Society, solicit and receive papers for consideration by the

Publication Committee, and select from the papers which have been approved by the Publication Committee those for presentation at technical sessions.

(*e*) *Regions Central Committee*, consisting of the Second Vice President and the seven (7) Regional Directors. The said committee shall consider and report to the Council on the activities of Chapters and Branches, and make recommendations to the Council concerning the policies, procedures and operation of the Society, its Chapters and Branches; it shall coordinate the activities of the Chapters Regional Committees; and it shall investigate applications for the creation of Chapters and Branches, and report thereon to the Council. The Second Vice President shall be the chairman of said committee.

Section 3. ***General Committees.*** Unless otherwise provided, the General Committees, and the respective Chairmen thereof, shall be appointed by the President, with the approval of the Council, as soon as practicable after the close of the Annual Meeting. The President, with the approval of the Council, shall at the same time appoint the chairman and the vice chairman of the Committee on Research. The chairman of said committee shall have had at least one (1) year of service on said committee. The vice chairman shall perform the duties of the chairman in the latter's absence, disability, resignation or death. The following shall be the General Committees and their respective duties:

(*a*) *Admission and Advancement Committee*, consisting of three (3) MEMBERS in good standing as MEMBERS for at least ten (10) years and having at least three (3) years of service on the Council or on General Committees, which shall receive and consider applications for admission or advancement to Member, Associate Member, or Affiliate grade, make diligent scrutiny and inquiry as to the character and qualifications of applicants, and report to the Council on the eligibility of each applicant for admission or advancement. The correspondence and proceedings of said committee shall be secret and confidential, and its correspondence concerning unsuccessful applicants shall be destroyed within a reasonable time.

(*b*) *Publication Committee*, consisting of three (3) MEMBERS. Subject to the direction of the Council the said committee shall formulate the editorial policies of the Society and for all of its publications. The chairman of the said committee may appoint sub-committees of one (1) or more members to review and report to the committee on the quality and appropriateness for publication of papers and bulletins intended for presentation or presented at Society meetings and the discussions thereof. In the performance of its functions the said committee and its sub-committees shall be subject to the following conditions: (*a*) That the data recommended for publication shall tend toward the professional education of the individual engineer; (*b*) that such data shall be free from commercial bias; and (*c*) that such data shall tend to advance for the public benefit the sciences relating to the arts of heating, ventilating, cooling or air conditioning, or the allied arts and sciences.

(*c*) *Guide Committee*, consisting of nine (9) MEMBERS, to serve for a term of one (1) year commencing November 1. The said committee, in harmony with the editorial policies of the Society, shall compile the text section of THE GUIDE. The chairman of the Committee on Research shall be an ex-officio member of the said committee.

(*d*) *Charter and By-Laws Committee*, consisting of three (3) MEMBERS, which shall consider all matters affecting the Charter and By-Laws, Rules and Regulations, and make recommendations thereon to the Council.

(*e*) *F. Paul Anderson Committee*, consisting of the First Vice President and four (4) MEMBERS, to serve for a term of one (1) year, which shall receive from the Council the announced conditions for the making of the subsequent annual award of the F. Paul Anderson Medal, select, solicit and carefully consider candidates for the award, and make nominations to the Council of the members complying with the announced conditions and deemed worthy to be recipients of the award. The decision to confer the award and all matters relating thereto shall be by a majority vote of all the members of the Council and in accordance with the presentation letter of Thornton Lewis.

(*f*) *Chapters Regional Committees*, each serving one Regional Area, and each consisting of the Regional Director for the Area and one (1) member and one (1) alternate member selected by each Chapter therein, to serve for a term of one (1) year. The said committees shall solicit from the Chapters and Branches within their respective Regional Areas recommendations concerning the policies, procedures, and operation of the Society, its Chapters and Branches, review the same, and make recommendations thereon to the Regions Central Committee. Said committees shall select the MEMBERS and alternates to serve on the Nominating Committee, and duly notify the Executive Secretary of such selections. The alternate members of Chapters Regional Committees may be present at committee meetings and participate in the deliberations thereof, but shall not vote therein except in the absence of the committee members for whom they respectively are alternates. The said committees shall hold committee meetings prior to June 1 of each year. Each Regional Director shall be the chairman of the Committee serving his Regional Area.

(*h*) *Committee on Research*, consisting of fifteen (15) MEMBERS, nominated by the Council or as provided in **ARTICLE VIII**, *Section 4*, and elected by the Society in the manner of elected officers. Subject to the direction of the Council, the said committee shall conduct and coordinate fundamental and basic research, and study and determine the principles and laws underlying matters in the sciences relating to the arts of heating, ventilating, cooling and air conditioning, and the allied arts and sciences, and cooperate with universities, colleges, schools and other organizations and groups, including governmental agencies, in the investigation of research subjects, subject to the proviso that all of the foregoing activities shall be devoted to the public welfare and general benefit, and shall not be designed to promote any individual, private or commercial interests. No contributor shall be specially favored on account of any contribution for research, which shall be used only for the public welfare. The said committee shall submit to the Publication Committee and to the Executive Secretary of the Society, its papers and reports concerning its investigations and activities.

There shall be a *Research Executive Committee*, consisting of the chairman, the vice chairman and three (3) other members of the Committee on Research appointed by the chairman, which shall exercise the powers and carry out the purpose of the Committee on Research during intervals between Committee meetings.

The chairman of the Committee on Research shall appoint such *Technical Advisory Committees* as may be deemed expedient to advise the Committee on Research and the Director of Research on specific research projects. At least one (1) member of each Technical Advisory Committee shall be a member of the Committee on Research and the chairman of the Committee on Research and the Director of Research shall be ex-officio members thereof. Technical Advisory Committees shall be governed by such rules and regulations as may be recommended by the Committee on Research and adopted by the Council.

The chairman of the Committee on Research may appoint a *Technical Adviser* as the Committee's consultant, who shall serve without compensation.

The Committee on Research shall recommend to the Council for appointment a *Director of Research*, whose activities, proceedings and reports shall be subject to the direction of the committee and the approval of the Council. The salary of the Director of Research shall be fixed by the Council upon the recommendation of the Committee on Research, and his employment may be terminated in the Council's discretion.

The Director of Research shall direct the operations of the Society's research program. Subject to the approval of the Committee on Research, the Director of Research may employ such assistants and other personnel as may be deemed by said committee to be necessary. The Director of Research shall be provided with a revolving fund in an amount fixed by the Council upon the recommendation of the Finance Committee, for the payment of the compensation of part time or temporary employees, travelling expenses and incidental petty cash outlays. He shall approve all purchase invoices and disbursements for research, and submit vouchers for the payment thereof at least semi-

monthly to the Executive Secretary. The Director of Research shall assist the Committee on Research in the preparation of its Budget for submission to the Finance Committee of the Society. The Director of Research and such other research assistants as may be employed in connection with research, shall, in consideration of such employment, agree in writing that any inventions, discoveries, ideas, plans, processes, formulae, experimental results and information received, published, divulged, made, or developed, or which may be in the course thereof while engaged in such employment, shall belong to the Society, and that no material relating thereto will be submitted elsewhere without the Council's consent.

(*i*) *Public Relations Committee*, consisting of a member of Council and three (3) members who shall serve a term of one (1) year. The said committee shall publicize the aims, activities and achievements, the scientific, and educational purposes of the Society, with the object of cultivating and stimulating public and members' interest in the Society and its affairs.

(*j*) *Standards Committee*, consisting of six (6) Members. The said Committee shall consider all scientific questions and data pertaining to engineering codes and standards, initiate and propose changes and improvements thereof for the public benefit, and report its recommendations to the Council.

Section 4. ***Other Committees.*** The Council may from time to time appoint other committees of one (1) or more members, and define their powers and duties, and it may abolish any such committees.

Section 5. ***General Provisions Concerning Committees.*** All Council, General and Other Committees of the Society, except the Nominating Committee, are subject to the following provisions:

(*a*) Except as otherwise provided, members on General Committees shall be divided into three (3) classes of equal number, each class to serve for three (3) years, and one-third of the prescribed number to be appointed, or in the case of the Committee on Research elected, annually.

(*b*) Except as otherwise provided, the committee year of all committees shall run from the close of the Annual Meeting to the close of the next Annual Meeting. The committee members whose terms expire shall continue in office until their respective successors are appointed or elected.

(*c*) Except as otherwise provided, each committee shall meet as soon as practicable after its appointment or election and may hold semi-annual meetings and such special meetings as the chairman or one-third of the members thereof may call. Each committee may recommend rules of procedure, which shall become operative upon the approval of the Council. Unless waived in writing or by telegraph or cable, notice of all semi-annual and special meetings of all committees shall be given in writing to all committee members and to the Executive Secretary not less than fifteen (15) nor more than thirty (30) days before the date fixed for the meeting. At committee meetings a majority of the members in office shall constitute a quorum. The committees shall promptly and fully report their activities to the Council and file with the Executive Secretary the minutes of their meetings and a complete record of their proceedings.

(*d*) The Council may by a two-thirds vote remove a member of any committee, and may, by a majority vote, designate a member to fill the vacancy or any other vacancy arising.

(*e*) Each committee shall submit to the Finance Committee on or before September 15 of each year an estimate of its disbursements for the ensuing fiscal year, and no committee, or member thereof, shall have the authority to incur any indebtedness or pecuniary obligation for which the Society shall be responsible, or claim reimbursement for advances, except to the extent authorized in the Budget or by resolution of the Council. Each committee member shall be responsible for the proper application of all funds remitted to him.

(*f*) Each committee's actions, proceedings, findings, conclusions and reports shall be subject to the direction and review of the Council, and the Council may take such steps, or see that such steps are taken by the committees as may be appropriate to comply with the Charter and By-Laws, and to make effective any resolution adopted by the Society or any resolution, rule or directive of the Council.

(*g*) If any doubt or controversy should arise as to whether a particular subject or matter is within the jurisdiction of a committee or whether any action should be taken by a committee, or in the case of a committee tie vote, the same shall be settled and determined by the Council.

(*h*) Except as otherwise provided, ex-officio members of committees may participate in committee deliberations but shall have no vote therein.

ARTICLE VIII

Meetings, Nominations and Elections

Section 1. ***Meetings.*** The Annual Meeting of the Society shall commence during the thirty (30)-day period beginning with the fourth Monday in January, and shall continue from day to day as the Council may arrange. Semi-Annual Meetings shall be held at such times as may be fixed by the Council. Special Meetings may be called at any time by the Council, and shall be called by the Council upon the written request of the President or of fifty (50) MEMBERS of the Society. Meetings shall be held at such place or places as the Council may designate, and shall be governed by Robert's Rules of Order, Revised, except when inconsistent with the laws of the State of New York, the Charter or these By-Laws. At any meeting of the Society, the presence of fifty (50) members entitled to vote shall be necessary to constitute a quorum.

Section 2. ***Notices.*** Notice of the Annual, Semi-Annual and of any Special Meeting of the Society shall be given in writing by the Executive Secretary and mailed, postage prepaid, not less than twenty (20) nor more than forty (40) days before the date fixed for the meeting, to each member of the Society at his last known address appearing on the records of the Society, and shall be published in the JOURNAL. Notices of Special Meetings shall state the purpose or purposes for which the meeting is called, and no business other than that set forth in the notice shall be entertained or transacted thereat.

Section 3. ***Nominating Committee.*** The Nominating Committee, to serve for a term of one (1) year from the opening of the Annual Meeting, shall consist of eleven (11) MEMBERS and sixteen (16) alternate MEMBERS, all of whom shall have been in good standing as MEMBERS for a period of at least five (5) years. Four (4) members and one (1) first alternate and one (1) second alternate member of the said committee, each from a different Regional Area, shall be selected by the Council at or prior to its last quarter-annual meeting; and one (1) member and one (1) first and one (1) second alternate member shall be selected by each Chapters Regional Committee, and certified to the Executive Secretary by September 1 of each year. No member of the Council shall be eligible to serve on the Nominating Committee, and no Chapter shall be represented on the said committee by more than one (1) of the eleven (11) MEMBERS. The Nominating Committee so chosen shall effect its own organization during the Annual Meeting of the Society, and shall hold a meeting during the Semi-Annual Meeting of the Society. By September 1 of each year the Nominating Committee shall nominate candidates for the elective offices, the members of the Council and the Regional Directors to be elected at the ensuing Annual Meeting, and notify the Executive Secretary of the names of the nominees, the notice to be accompanied by the written acceptances of the candidates. Seven (7) members entitled to vote on the said committee shall constitute a quorum. No alternate member may be present at committee meetings, nor shall he participate in the deliberations thereof or vote therein except in

the absence of the member, or the member and the first alternate member of the one of the eight (8) groups with which he was selected. The transportation expenses, as defined by the Council, of eleven (11) committee members who participate in the deliberations at the annual and semi-annual meetings of said committee shall be included in the budget and defrayed by the Society.

Section 4. ***Other Nominations.*** Nominations of officers and members of the Council, other than those nominated by the Nominating Committee, and nominations of members of the Committee on Research, other than those nominated by the Council, may be made in writing by at least fifty (50) members eligible to vote, upon presentation of such nominations, with each nominee's consent, to the Executive Secretary at least sixty (60) days prior to the opening of the first session of the Annual Meeting, whereupon the nominees' names shall be placed upon the ballot with a notation that they are presented by members independent of the Nominating Committee.

Section 5. ***Voting.*** Voting at any meeting may be in person or by proxy, but only the Executive Secretary and MEMBERS of the Society shall be eligible to act as proxies. Proxies shall not be valid for more than three (3) months from dates of execution. The Executive Secretary and the MEMBERS acting as proxies shall hold the ballots of their principals secret and confidential. Voting for election of officers, Council members, members of the Committee on Research, on proposals to amend these By-Laws, and on questions required to be referred to the Society pursuant to **ARTICLE V**, *Section 2*, shall be by secret ballot. In the event of any tie vote, the Council shall decide the vote.

Section 6. ***Ballots.*** Together with notice of the Annual Meeting, the Executive Secretary shall forward appropriate proxies and ballots to members entitled to vote. The proxies and ballots shall contain spaces for write-in names.

Section 7. ***Results.*** The polls for election shall be opened at the opening of the Annual Meeting and shall remain open for a period of five (5) hours. Thereafter the ballots shall be opened by three (3) inspectors of election appointed by the President, who shall be authorized to fill any vacancy occurring among such inspectors. The inspectors of election shall consider ballots and votes to be valid provided the intent of the voter is clear. The result of the vote shall be reported by the inspectors of election in writing, and shall be announced by the President on the second day of the Annual Meeting, whereupon the terms of the inspectors of election shall expire. The elected candidates shall be installed during the Annual Meeting and their terms shall commence at the close of the last session of the Annual Meeting.

ARTICLE IX

Amendments

Section 1. ***Prerequisites.*** These By-Laws may be amended by a two-thirds vote of the Society at an Annual Meeting thereof, provided that written notice of the proposed amendment, subscribed by two-thirds of the members of the Council or by fifty (50) MEMBERS, be given at a previous stated or Special Meeting, and that notice thereof as pertinently amended by majority vote at said stated or Special Meeting be also given by the Executive Secretary in the notice of the Annual Meeting.

Section 2. ***Renumbering.*** The Council may, by a two-thirds vote, renumber existing Articles or Sections of these By-Laws.

ARTICLE X

Adoption

Section 1. ***Effect.*** These By-Laws shall supersede all previous Constitutions, By-Laws and Rules of the Society, and shall come into effect upon the adjournment of the meeting at which these By-Laws shall be adopted.

1557

No. 1557

NOISE PRODUCTION AND DAMPING IN WATER PIPING

By W. L. ROGERS*, EVANSTON, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with Northwestern University, Evanston, Ill.

THE PRIMARY purpose of this progress report is to summarize the activity and developments of the past year in the project. Two previous reports[1, 2] have summarized the preliminary planning, some of the experimental apparatus and instrumentation, and experimentally-determined sound-pressure levels and frequencies produced by flow of cold water over a range of velocities through 1-in. pipes and fittings, including orifices. The information presented here should not be viewed as constituting a terminal report on any aspect of the problems under investigation, but only as a report of progress.

The basic instrumentation and procedures are essentially like those discussed in the 1954 report on this project. All sound measurements were made by means of a microphone mounted through a hole in a slightly flattened area of the pipe wall, essentially flush with the inside surface. Thus the noise *in the water* was measured. Such a procedure allows evaluation of pipes and fittings as *noise-producers*, leaving the manners in which this noise is amplified, transmitted, damped, or becomes airborne as separate problems. The total problem thus is divided into its natural subdivisions.

The decibel unit used throughout this report is the same as that used in the 1954 report[2], and is defined as

$$\text{db} = 20 \log_{10} p$$

in which p is the sound pressure expressed in dynes per square centimeter. Most of the measurements reported can be considered accurate within 1 or 2 db, the amount of variation depending upon the magnitude and upon the extent of fluctuation of the sounds measured.

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* Assistant Professor of Mechanical Engineering, Northwestern University.

[1] Exponent numerals refer to References.

Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

FIG. 1. POSITION OF MICROPHONE IN EACH OF THE TEST LENGTHS OF 1½-IN. PIPE. FOR TESTS OF FITTINGS, EACH FITTING OCCUPIED THE POSITION LABELLED ORIFICE

WATER FLOW-NOISE IN 1½-IN. STRAIGHT PIPE

Fig. 1 shows the position of the microphone in each of the 2 lengths of 1½-in. pipe tested. The 2 total lengths were 9.78 ft and 27.04 ft. The additional 1-ft length shown was used only when testing orifices and other fittings, and not during the straight-pipe runs. The inlet and discharge lines connected to the test lengths contained suitable sound-damping arrangements to prevent transmission of control-valve noises into the test lengths.

Fig. 2. shows the microphone mounting. Considerable care was used in selecting the unions and in minimizing changes in section inside them. Evident in the photo-

FIG. 2. MICROPHONE MOUNTING ARRANGEMENT IN 1½-IN. PIPE

graph is the thin rubber membrane used to cover the flat microphone face. A thin layer of silicone oil between the membrane and the microphone face insured the absence of trapped air which could affect the microphone readings.

Fig. 3 presents the experimental measurements determined for cold water (38 to 70 F) and for inlet pressure of 20 to 25 psig. Fig. 4 was derived from Fig. 3, and the

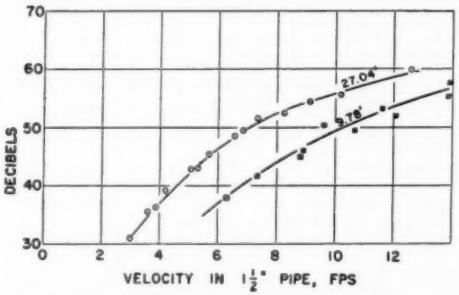

FIG. 3. SOUND-PRESSURE LEVELS PRODUCED BY FLOW OF WATER THROUGH TWO LENGTHS OF 1½-IN. STRAIGHT PIPE

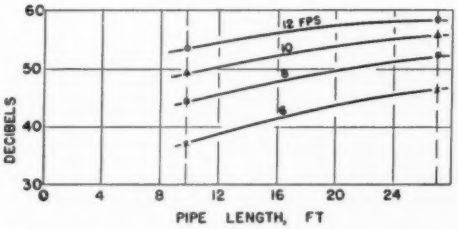

FIG. 4. EFFECT OF PIPE LENGTH UPON FLOW NOISE GENERATED IN 1½-IN. STRAIGHT PIPE AT VARIOUS VELOCITIES

curves were faired in to correspond to the general trends of the similar curves for 1-in. pipe presented in the earlier report[2]. The trends are correct, although there may be some error in the curves, as each was drawn through only 2 plotted points.

CORRELATION OF STRAIGHT-PIPE FLOW-NOISE MEASUREMENTS

Fig. 5 presents the most satisfactory correlation of straight-pipe flow-noise data obtained to date. The sound pressure produced per sq in. of pipe flow area is plotted as a function of pressure drop per foot of pipe due to flow. In other words, the intensity of noise production is a function of the intensity of pressure drop. The relationship appears to be approximately a linear one, when sound pressure in dynes per square centimeter (*not* in decibel units) is used as the ordinate. Only the longer lengths of 1-in. and 1½-in. pipes tested have been plotted. For the shorter lengths

not plotted, considerably less than 100 pipe diameters were present before the microphone, and it is felt that possibly the full equilibrium turbulence pattern was not established in the water by the time the water reached the microphone location. Points for the shorter lengths fell below the curves of Fig. 5, lending credence to this thought. Also, the higher sound levels produced in the longer pipes were capable of more accurate measurement.

The pressure drops of Fig. 5 were *calculated*, using a plot of the Colebrook function† to determine friction factors. Reynolds numbers could be calculated accurately, but it was necessary to assume values of pipe surface roughness before friction factors could be determined. Values given for new wrought iron and steel pipe were used, although pipes tested were not all new, and even the new pipes unquestionably changed somewhat before all tests were completed. The 2 lines drawn

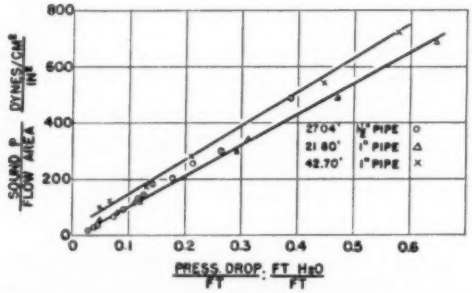

FIG. 5. CORRELATION OF NOISES PRODUCED BY FLOW OF WATER THROUGH STRAIGHT PIPES

in Fig. 5, and the spread of the plotted points, may well reflect only the effects of the uncertainties concerning the relative roughnesses and friction factors. With this in mind, the correlation appears to be reasonably satisfactory.

Actual pressure drops were not measured because of the possibility that pressure taps might contribute unwanted noise or otherwise affect the sound levels being measured.

In 1½-in. pipe, the predominant frequencies produced were similar to those previously determined in 1-in. pipe. The band from 20 to 75 cycles per second (cps) usually contained the major part of the total signal, with significant amounts lying also in the 75–150 cps band, and occasionally in the 150–300 cps band. Above 300 cps, there was negligible contribution to the total signal. Occasionally there were hints that selective amplification of certain of the frequencies present could occur by resonance in the piping circuit. For example, in some runs a note was heard in the monitoring earphones, and the same frequency could be reinforced by thumping a rubber hose used in the feed line. The contribution of the note to the total noise level was not large (although not negligible either) but the note was discernible. In general, however, the noises were broad-band in character, as they were also for 1-in. pipe.

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† Mechanical Engineers' Handbook, 5th Edition, by L. S. Marks, Editor-in-Chief (McGraw-Hill Book Co., Inc., New York, N. Y.).

In a series of check runs using 1-in. pipe, water temperature was varied over a range from 37 to 129 F. Use of the microphone at temperatures higher than this was not deemed advisable. The data showed a slight decrease in noise produced at the higher temperatures—a maximum reduction of approximately 2 db. Water velocities used in tests were 5 to 13 fps. The indications are that water temperature has only a small effect on flow-noise production in straight pipe as long as the water remains liquid. Conditions promoting cavitation during flow would be expected to promote intense noise generation simultaneously.

Although only 2 pipe sizes have been investigated, there is little reason to expect that other sizes would behave differently. Noise produced by fittings has been found to be considerably more significant than that produced in straight pipe, and this observation could be expected to be true for sizes not tested, also. If this is accepted

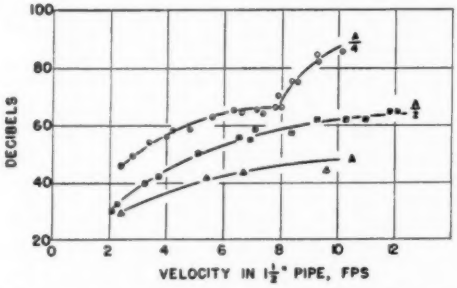

FIG. 6. SOUND-PRESSURE LEVELS PRODUCED BY FLOW THROUGH ORIFICES

as true, and because as flow velocity is increased through straight pipe it must also necessarily be increased through the fittings of a system, it is concluded that noise produced by flow of liquid water through straight pipe must be assigned to a position of secondary importance most of the time. For very long, straight runs with few fittings or in pipes not flowing full, this conclusion might have to be modified.

FLOW-NOISE THROUGH ORIFICES

Orifices in 1½-in. Pipe: Noise levels produced by flow of water through orifices having ¼ and ½ of the full flow area of 1½-in. pipe are shown in Fig. 6. The tests were made with cold water, with inlet pressure of 15 to 20 psig. The trends are the same as those found in 1-in. pipe. The abrupt increase in noise for the A/4 orifice is attributed to cavitation. At velocities higher than those of the test, the A/2 curve should exhibit the same characteristic.

Fig. 7 is derived from Fig. 6 and illustrates the effect of orifice size on the production of noise at various flow velocities. Points involving cavitation are *not* included in this curve.

The trend of frequencies was the same as for 1-in. pipe—a rise in importance of the higher frequencies at the higher flow rates. At low flow rates (say 4 fps) frequencies of importance were mostly in the 150–600 cps range. After the onset of cavitation, frequencies of importance extended up to and beyond 2400 cps.

Effect of Temperature: Fig. 8 shows the results of tests made at 2 velocities for the A/4 orifice in 1½-in. pipe. Temperature of the water was varied between about 50 and 130 F. From Fig. 6 it can be seen that the run at the higher velocity (9.2 fps) occurred with conditions giving intense cavitation, while there was no cavitation at all in the lower-speed run (at 4.0 fps). In either instance, water temperature within the range of the tests had little effect on the production of noise. Without cavita-

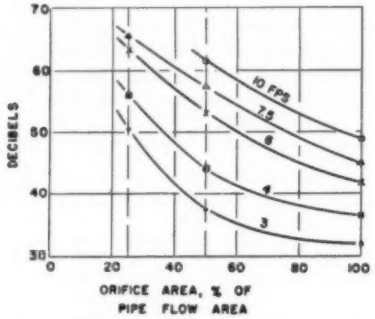

FIG. 7. EFFECT OF ORIFICE AREA UPON NOISE GENERATION FOR FLOW AT VARIOUS VELOCITIES IN 1½-IN. APPROACH PIPE

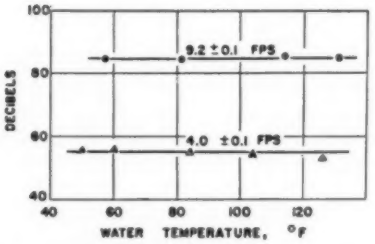

FIG. 8. EFFECT OF WATER TEMPERATURE UPON FLOW-NOISE PRODUCTION OF THE A/4 ORIFICE IN 1½-IN. PIPE

tion, there appears to be a slight tendency toward reduced noise at higher temperature, as was noted also for straight pipe. The change appears to be inconsequential, however (within the range of the tests).

In check runs in 1-in. pipe, water at 127-128 F was run through an A/8 orifice at various velocities, and the points fell on the same curve of db *vs.* velocity as the points for cold water runs. It appears that for *liquid* water, noise produced by flow through orifices is not particularly sensitive to water temperature. This was indicated to be true also for straight-pipe flow. In addition, after full cavitation is

developed, the temperature again appears to exert an insignificant effect. However, the velocity at incipient cavitation (and the sudden noise increase) should be expected to be influenced by water temperature. At higher temperatures, cavitation should occur at lower velocities (at the same pressure).

Correlation of Flow-Noise Measurements: Fig. 9 was plotted in an attempt to relate sound pressure levels to abruptness of pressure change across orifices. All points represent runs made at velocities below those at which cavitation began. The pressure drop was *calculated*, not measured. It was considered inadvisable to use pressure taps because their effect on the noise production of the system was unknown. For the calculated pressure differences, flow coefficients for sharp-edged orifices were taken from the *ASME* publication *Fluid Meters*, Part I, 4th Edition, 1937.

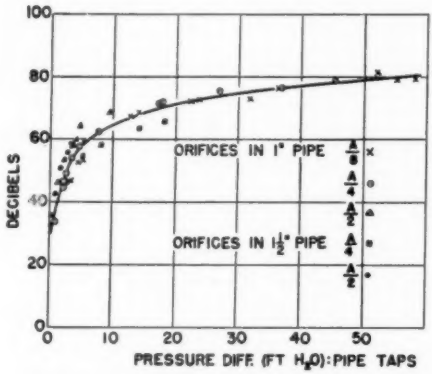

FIG. 9. CORRELATION OF FLOW NOISES THROUGH ORIFICES

Experimental measurements gave the actual flow rate in each run and allowed the calculation of Reynolds number. In the tables for *pipe taps*, the flow coefficient k was listed as a function of Reynolds number and the ratio of diameter of orifice to diameter of pipe. With the flow coefficient and the actual flow rate known, the pressure drop for pipe taps was calculated from equation:

$$q = A_{\text{orifice}} (2\,gh)^{1/2} (k)$$

in which

q = cubic feet per second of actual flow.
h = pressure drop, feet of water.
k = flow coefficient.

Pipe tap locations, as indicated in the *Fluid Meters* publication, are at 1 or 2 pipe diameters upstream from the orifice, and at 4 to 8 pipe diameters downstream at or beyond the location of the maximum downstream static pressure. Values of k for 2-in. pipe were used because smaller pipe sizes are not listed. However, comparison of values given for 2, 3, 4, and 6-in. pipes indicate that the error involved should be very small.

The correlation indicated by Fig. 9 is not perfect, but it is evident that pressure drop is a significant factor. For any one orifice, all points lie reasonably well on one curve, and the curves for different orifices lie rather close together. When it is considered that the condition of the supposedly sharp edge of an orifice exerts a marked influence on the flow rate and pressure drop, and probably also on the production of noise, it is not inconceivable that the differences in the plotted points might reflect the relative sharpness or roundness of the orifice edges. With this thought in mind, the correlation may be much better than a first glance might indicate. The orifices tested were turned on a lathe with reasonable care, but there was no meticulous inspection of edge sharpness and no use of microscopic examination.

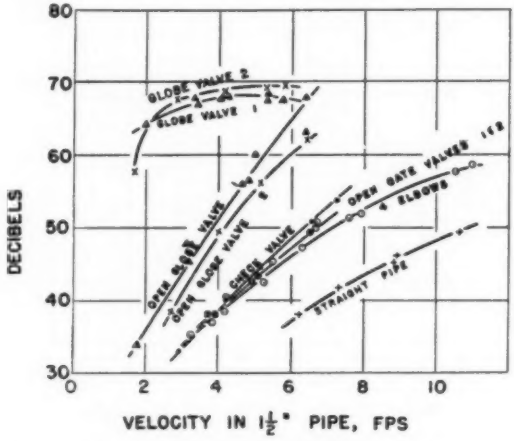

FIG. 10. SOUND-PRESSURE LEVELS PRODUCED BY FLOW THROUGH 1½-IN. FITTINGS

It is interesting to note that a plot of the average curve shown in Fig. 9 is approximately a straight line when the sound pressure is expressed in dynes per square centimeter. Thus, if the pressure drop is doubled, the sound pressure expressed in dynes per square centimeter (not in decibels) is approximately doubled also. This assumes no cavitation, which would cause a much greater increase. The same linear trend is evident when each orifice is plotted separately.

FLOW NOISE THROUGH 1½-INCH VALVES AND ELBOWS

Fig. 10 indicates the relative importance of several typical 1½-in. fittings as noise producers. The trends and orders of magnitude are essentially the same as those reported previously for 1-in. fittings. The highest noise levels are produced by the globe valves used to throttle the flow. Two samples were checked. Fully-open globe valves are considerably noisier than open gate valves. The 4 elbows were arranged in a U shape with 2-ft sides and bottom. They cause as much noise as a considerable length of straight pipe.

PRELIMINARY PUMP TESTS

With a collection of information available on noises produced by flow of water through pipes and fittings, it was logical next to compare this information with the magnitudes of noise produced in the water by pumps. A typical 1-in. circulator of the type used in heating systems was selected for some preliminary experimentation. It was arranged to operate in a closed system, taking water from a supply tank, discharging it into the test section containing the microphone, from which the water returned to the tank. Suitable sound-damping arrangements preceded and followed the pump and microphone section. The system pressure and temperature could be regulated to desired values.

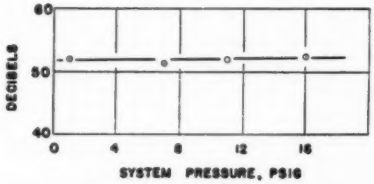

FIG. 11. EFFECT OF SYSTEM PRESSURE UPON PUMP NOISE PRODUCED IN THE WATER

In Fig. 11 are plotted the results of a run made at different levels of system pressure, this pressure being measured a short distance upstream from the pump. Flow velocity in the 1-in. pipe was 1.08 fps for all runs, a somewhat low value for the pump, produced by causing the pump to discharge against a relatively high pressure. Water temperature was 83 F. The order of magnitude of the noise was about the same as that of many of the fittings already discussed. System pressure did not affect the amount of noise generated.

A series of runs was made also at constant system pressure of about 12 psig (before the pump) and for a range of velocities from zero to about 6 fps. There was no large effect of velocity on sound produced, a total spread of about 2 db being found.

The top curve of Fig. 12 indicates that water temperature, too, had an inconsequential effect within the range tested (55 — 130 F).

Generalizations cannot be made from tests of 1 pump operated within limited ranges of variables. However, it can be stated that *flow* noises *in the water* obviously can be as large or larger than *pump* noises produced in the water. The pump, however, constitutes a greater noise hazard because its noise is concentrated at certain frequencies. Thus if a natural frequency of the liquid circuit or of the pipe structure and hanging arrangement coincides with one of the pump's predominant frequencies, there is available a great deal more input energy to excite vibration and noise than is available in the relatively broadband noises produced by fittings, in general.

In addition to the noise generated in the water (and this is transferred rather readily to the pipe wall) the pump and motor can cause vibration of the pipe wall directly by contact. This vibration reaches the water in the pipe rather readily, too. The pipe wall and water inside are more closely coupled than if the pipe contained a gas. For mechanical isolation of the pump and motor vibrations from the pipe wall,

the basic principles are almost self-evident: soft mountings and connections with natural frequencies low relative to disturbing frequencies. This aspect of the total problem has not received attention in the current study because it is felt that more is unknown about certain other aspects.

SOUND TRANSMISSION ALONG THE WATER PATH

At the conclusion of the pump tests, it was decided to evaluate an experimental sound damping chamber by placing it between the pump discharge and the microphone. The chamber consisted of a 12-in. length of 6-in. pipe packed full of small-bore glass tubes having outside diameter of 4 mm each and wall thickness of approxi-

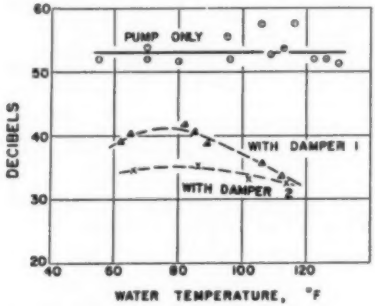

FIG. 12. EFFECT OF WATER TEMPERATURE ON PUMP NOISE PRODUCED IN THE WATER, AND THE EFFECTIVENESS OF 2 EXPERIMENTAL SOUND DAMPERS IN REDUCING LIQUID-PATH TRANSMISSION OF PUMP NOISE

mately 0.8 mm. The small tube size was expected to effectively damp pressure oscillations by viscous drag. The transitions from 1-in. pipe to 6-in. pipe were accomplished with standard fittings. At the pump-discharge end the transition was 21 in. long, and at the microphone end it was only 8 in. long. This damper is labelled No. 2 in Fig. 12, and it is obvious that it performed very satisfactorily.

Next it was decided that a sudden enlargement with no filling should be tried. From theoretical considerations, it was anticipated that such an enlargement should act as a low-pass filter, cutting off frequencies above a certain value. The exact effect on the cut-off frequency of variations in area ratio, volume, and other factors was not known. A chamber readily available had an inside diameter of 7¾ in. and a length of 17½ in. This chamber was used, with results as indicated by the curve for damper No. 1 in Fig. 12. The results were encouraging, the sudden enlargement seemed to offer a simple and practical means of blocking transmission along the water path, and so it was decided to explore the variables involved.

The author knows of instances in which he has been convinced that transmission of noise along the water path has been at the heart of the trouble. In other instances, resonance within the liquid circuit seemed to be a possibility. In such circumstances,

the use of sudden enlargements as filters should be capable of solving the problem and has solved it in some actual installations. Quantitative information for design purposes is needed, however.

EXPERIMENTAL APPARATUS FOR EVALUATING DAMPER PERFORMANCE

It was decided to assemble a system containing a source of noise of controlled frequencies and amplitudes which could be fed into the liquid water in a pipe. Fig. 13 shows the system which was evolved after considerable experimentation. The vertical feature allows the 1-in. pipe to be filled from the bottom with a minimum tendency to trap air, which caused considerable trouble in earlier horizontal runs. The pressure in the system can be reproduced readily, and also the length of water column.

The bottom of the pipe is covered with a rubber membrane fastened with a hose clamp. At the center of the membrane a small circular flat plate serves as a piston. Attached to this plate is a bolt which also is attached to the vibration motor. Vertical

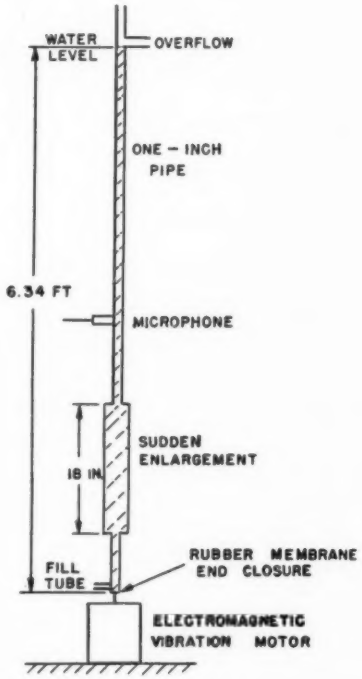

FIG. 13. EXPERIMENTAL ARRANGEMENT FOR EVALUATING LIQUID-PATH SOUND DAMPERS OR FILTERS

oscillations of the armature of the vibration motor thus produce vertical piston motion and corresponding pressure pulsations in the water.

In all runs, the exciter coil of the vibration motor was supplied with the same constant d-c voltage, and the armature (moving coil) with the same constant amperage.

EXPERIMENTAL DATA—THREE DAMPER SIZES

The system of Fig. 13 is open at the top, and so it was expected that the fundamental natural frequency of vertical wave travel in the water in the pipe would correspond to that for which the length of water column is a half wavelength. The frequency would be, then, the speed of sound in the pipe divided by 2(6.34). The speed of sound in a pipe is reduced by the expansion of the pipe walls, which cannot be assumed to be perfectly rigid when a pipe contains water. The effect can be calculated. Ganitta's equation (which was used also in Reference 3) gives the ratio of the speed of sound c in a pipe to the speed c_0 in free water as

$$\frac{c}{c_o} = \left[\frac{1}{1 + \dfrac{2\rho c_o^2 d}{g\left(1 - \dfrac{5a}{6d}\right) aE}} \right]^{\frac{1}{2}}$$

in which

ρ = water density.
g = gravitational constant.
d = inside pipe diameter.
a = pipe wall thickness.
E = Young's modulus of pipe material.

For standard 1-in. pipe, $c/c_0 = 1/1.09$. Using a value of 4,794 fps for c_0, c = 4,390 fps. Therefore, the calculated fundamental resonant frequency for wave travel in the water column of Fig. 13 is 4,390/12.68, or 346 cps.

Experimentally, the resonant frequency was found to be 343 cps, as indicated by the peak of the top curve of Fig. 14. The data for this top curve were obtained with 1-in. pipe used in place of the sudden enlargement shown in Fig. 13. The resonant peak below 100 cps is attributed, at the moment and pending fuller investigation, to *structural* resonance of the pipe. Draining the water caused a marked drop in microphone output although to the ear the sound level remained unchanged. That there is much poorer transfer of pipe-wall vibration to gas than to liquid is a known fact, and so there appears to be substantiation for the assumption of structural resonance.

The characteristics of 3 sizes of sudden enlargements used as sound dampers are shown in Fig. 14. The 3 standard pipe sizes all had lengths of 18 in. The change in section was sudden, produced by tapping flat end plates for 1-in. pipe. The endplates were welded to the pipes.

It is apparent that the enlargements are effective, and that the larger sizes become effective at lower frequencies. This is the anticipated characteristic of low-pass filter action. It seems, further, that the required sizes of enlargements are not prohibitively large, and that they offer a practical means of minimizing sound transmission along the water path when this is a problem. The data of Fig. 14 also support the proposition that resonance within the water column of a piping system can intensify or create a noise problem which might be of little consequence in the absence

of the resonant condition, and that a sudden enlargement can cure such a problem. Although Fig. 14 extends only to 600 cps, the actual tests were run to 2600 cps. With straight pipe, higher harmonics were excited, but with the filters in use, there was no microphone signal beyond those plotted in the figure.

The quantitative evaluation of the factors of importance to the filter action is continuing. The data of Fig. 14 indicate the feasibility of the filters, but a more complete quantitative report must wait upon additional testing. The volume of the enlarge-

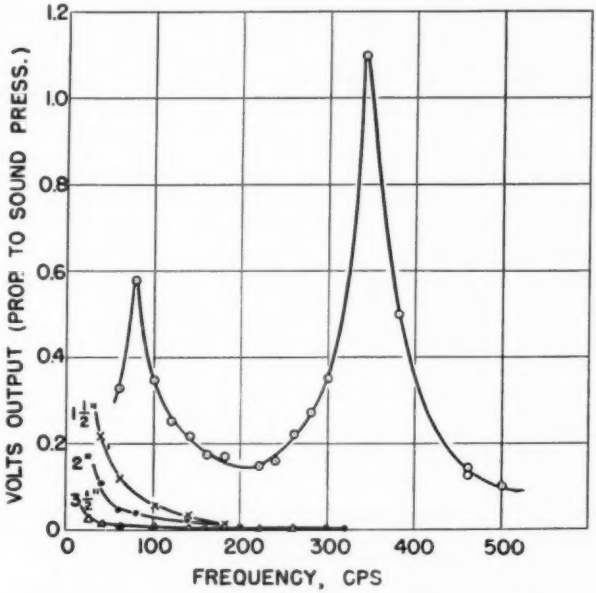

FIG. 14. EFFECTIVENESS AND CHARACTERISTICS OF 3 SIZES OF SUDDEN ENLARGEMENTS IN 1-IN. PIPE AS ACOUSTIC LOW-PASS FILTERS

ment is expected to be of considerable importance, theoretical considerations indicating that the larger the volume, the lower the cut-off frequency.

Other logical future activity should include consideration of high-pass filters as well as the low-pass type. Depending upon the situation, one type might be preferable to the other. A study of the phenomenon of resonance in the water path of a closed piping system containing a source of noise should also be capable of contributing to understanding of the ways in which a noise problem can appear in a practical system.

ACKNOWLEDGMENT

The author wishes to express appreciation to Messrs. Duclos, Grenleski, Grounds, Huellmantel, and Ross, all Graduate Assistants at Northwestern University. Each of

these men assisted in some measure in the activity of this project during the past year. The continuing cooperation of Prof. B. H. Jennings is also acknowledged.

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DISCUSSION

F. H. FAUST, Bloomfield, N. J.: What is the principle of quieting the noise through these glass tubes? Does the change of velocity in the tubes contribute to quieting the noise?

PROFESSOR B. J. JENNINGS: I am sorry that I am no sound expert. I wish Professor Rogers were here to answer these questions. But it is my surmise in this connection that it is similar to something that the people in ductwork have known for a long time, that is, if one can distribute the sound-absorbing area, some of the noise which exists is absorbed and damped out. For example, if one takes a duct carrying air in an air-conditioning system and puts in a honeycomb passage, a large amount of the noise is absorbed or damped out.

L. N. HUNTER, Johnstown, Pa.: These curves indicate that temperature and pressure apparently had no effect upon the noise. I would like to ask Professor Jennings if he feels that these tests covered a wide enough range to indicate that temperature and pressure do not have an effect on the point at which cavitation starts.

PROFESSOR JENNINGS: As temperature increases one must have a higher pressure on the system to keep evaporation from occurring. I judge from these curves that as long as you stay below the point of boiling, the temperature-pressure relationship does not seem to be critical to noise in the system. As temperature increases you approach a point close to saturation pressure and would expect to find that the tendency toward cavitation is increased greatly.

F. J. LINSENMEYER, Detroit, Mich.: There are 2 questions I would like to ask. They are perhaps covered in the paper, but I haven't had a chance to read it. One is the possible relationship between energy within the pipe or within the water, you might say, and the effect that you get outside—in other words, the avoidable noise—and what relationship there may be from curves.

The second one is the effect of entrained air in the water. One generally finds that when a control valve is noisy, it is frequently caused by air in the system. I think in most cases the noise can be overcome by freeing whatever air may be carried around in the main.

PROFESSOR JENNINGS: Noise is very closely related to the question of air in the system whenever the air is released at an obstruction point, such as the edge of a valve or at a control point.

Under normal conditions, such as might exist in a straight pipe, there is usually no obstruction to cause air release or cavitation to occur. In valves which are well open, again there is little tendency for this to occur.

As a valve is partly closed and a really significant pressure drop occurs across it, when

there is air in the system it is very readily released and under those conditions noise occurs. If you remember, one of the graphs we presented was one which came up smoothly and then broke off very suddenly. That showed cavitation, air release or the like. We can get cavitation in 2 ways. One is by gas release within the water itself. The other is when vapor pressure equilibrium conditions are approached. The water is then hot compared to the pressure which exists on it, and boiling is independent of the air that is in the system. However, air is the first and most usual contributing factor to this noise which exists in the system.

HERBERT KREISMAN, Chicago, Ill.: I would like to ask Professor Jennings what consideration has been given to the relation between the diameter of the sound-absorbing-pipe and its length, and also its location in the straight pipe, whether it should be a certain distance from any elbow or tee; and secondly whether any consideration has been given in the experimentations to the mechanical noise of the pump, the actual physical transmission of vibration along the piping.

PROFESSOR JENNINGS: A great deal of thought has been given to this in the design and layout of the arrangements. It is very definitely true that if you have a pump which is externally vibrating and noisy, it can cause the pipes themselves to vibrate, and such vibration is purely a physical phenomenon, external to the liquid system, which is transported to another part of the building. As far as these investigations were concerned, Professor Rogers was mainly thinking of what happened inside the water system.

The first part of your question concerned the length of the sound-absorbing chamber, and I think I can safely say that that length is not so critical as is volume. For example, we would probably get along just as well, again thinking about 1-inch pipe, if we had a sound-absorbing chamber which was of the order of magnitude of a foot in length as we would if we had one that was 2 feet in length.

Your other question as to how far the absorber might be located away from an elbow would be only a guess on my part, and I don't think I will make it.

G. A. LINSKIE, Dallas, Texas: A short time ago we were mechanical contractors on a 9-story building in which the condenser water pumps were located in the basement, and the cooling tower located on the roof. During the operation of the air conditioning system, we experienced a high frequency noise throughout the building occurring when the 2 condenser water pumps, installed in parallel, were in operation.

The odd thing about this particular noise was that it occurred on the third, fifth, seventh, and ninth floors. By this I mean the intensity was greater on these floors than on the intermediate floors. Our problem was to locate, isolate, and dampen out the noise condition. Our first clue came when we found that the noise occurred only when both condenser water pumps were operating simultaneously.

We did realize that we had reduced the size of both the supply and discharge headers between pumps #1 and #2. In an effort to reduce the noise we disassembled both headers and installed new headers on both pumps, and at the same time increased the headers by possibly one or two pipe sizes. The connections in and out of the headers were made with long sweep tube turn fittings rather than the stub in connections originally installed. In a further effort to reduce the noise, we isolated the pumps with short rubber flexible connections, to stop any transmission up through the piping system.

The noise was very objectionable throughout the building and we were cutting and trying, so to speak, in order to find a solution. Our down time was limited, and it was necessary for us to piecemeal the changes in order to get the system back in operation as soon as possible. Over the week-end we had been able to increase the size of the supply and discharge header and install the flexible connections on both the suction and discharge outlets of each pump. The system was placed back in operation Monday morning, and much to our surprise the noise had disappeared.

We were unable to ascertain immediately what corrective measures eliminated the noise, but evidently, from the paper given here today, we must have put a "sound trap" in the system by increasing the header size and thereby interrupting the constant velocity

of the water system. We understand these noise problems are the exception rather than the rule, but when they do occur they can be very costly and annoying.

I think Mr. Roger's paper can be very helpful in solving similar noises, and being contractors, we are very interested in solving them at the lowest possible cost.

PROFESSOR JENNINGS: The finding of what is causing the disturbance in a system requires the ingenuity of a Sherlock Holmes, and it is luck as well as engineering. I think you were lucky.

AUTHOR'S CLOSURE (Professor W. L. Rogers) WRITTEN: In answer to Mr. Faust's question, the use of small tubes (not necessarily of glass) furnishes a large amount of surface area in contact with the flowing stream. The velocity of a flowing fluid is always zero at a solid boundary, and increases rapidly as we move away from that boundary and into the main stream. In the thin layer near the boundary, shear rates are high, and fluid pulsations must produce more shear when there is a large amount of enclosing boundary. Thus the pulsations are resisted by the accompanying shearing action, with a reduction in energy of the pulsations as shear occurs. This is, effectively, the principle of sound-absorption materials. Thus, even with the same total cross-sectional area for flow and with the same average velocity, increased sound absorption is to be expected from the use of small tubes.

In response to Mr. Hunter, it is obvious that temperature and pressure do have an effect on the velocity at which cavitation occurs in any flow passage. For a given pressure and temperature of liquid entering a flow component (say a valve), if the flow rate is fast enough, then at some localized region the pressure may drop to a low enough value to allow rapid escape of dissolved gas or vaporization of liquid, probably with subsequent reabsorption or condensation in downstream regions of increased pressure. These occurrences result in relatively intense noise generation. From their very nature, they are influenced by the pressure and temperature of the stream. Thus, in Fig. 6 for example, the onset of the phenomena would be expected to shift from the 8 fps point of the top curve for different pressures and temperatures. The test data indicate, however, relative insensitivity over a considerable range of temperature and pressure. For entering water nearer a boiling condition, a considerable shift to a lower velocity would be expected.

To Mr. Linsenmeyer, it is remarked that the transfer of noise in the water to vibration of the pipe wall, which creates audible noise, has not been studied intensively in this investigation. Reference 3 listed at the end of the paper deals with some aspects of this problem. In general, natural frequencies of the pipe (which are influenced by the manner of support) are easily excited by noise in the water.

Entrained air certainly is a contributing factor to noise. In fact, it is to be expected that liberation of entrained gas usually occurs at a lower velocity than that at which actual vaporization of the liquid occurs.

To answer Mr. Kreisman, and as indicated on the last page of the paper, further quantitative evaluation of the effects of length, diameter, and like factors, of the enlargements is needed. At the moment, an enlightened guess might be that *volume* of the enlargement is of considerable importance in establishing the cut-off frequency, and that placement near elbows and tees would not be a particularly critical factor.

The mechanical noise from a pump, and its transmission, have both received thought during the operation of this project. No experimentation has been performed as yet on this aspect of the total noise problem because it was felt that less was known and understood about certain other aspects.

We were glad to learn of Mr. Linskie's interesting experience. It is difficult to say which of his corrective measures solved his particular problem.

In closing, the author regrets that he was unable to be present personally at the Cincinnati meeting, and wishes to express his appreciation to Professor B. H. Jennings for his presentation of this report.

1558

No. 1558

PERFORMANCE OF COVERED HOT WATER FLOOR PANELS

Part 1—Thermal Characteristics

By E. L. SARTAIN* AND W. S. HARRIS**, URBANA, ILL.

IN 1951, the Floor Slab Laboratory, described in detail in a previous paper[1], was modified to facilitate the investigation of hot-water panel-heating using ¾-in. welded steel pipe embedded in concrete floor slabs. The Laboratory contained 4 test Rooms A, B, C and D with different types of floor insulation (see Reference 1, Fig. 3).

Heating System: The same hot-water, floor-panel heating-systems with bare panels[1] were used for tests with floor coverings. The panels were designed in accordance with the $I = B = R$ Installation Guide No. 6. In each room, hot water from the heater was supplied to the panel at the outside wall. Each test room was provided with its own electric water heater, watt-hour meters, circulating pump, necessary piping and controls.

Floor Coverings: At the beginning of the 1952–53 heating season rubber tile was installed on the floor of Room A and asphalt tile in Room B. Both were 9 x 9 x ⅛ in. squares and installed by a professional tile setter following the manufacturer's recommended methods, and a coat of wax was applied. Two types of carpeting and 3 pads were used (see manufacturers' specifications, Table 1). These carpets and pads were cut to room size and installed by professional men. Carpets and pads were selected in consultation with members of the Carpet Institute, Inc., as being representative of carpeting commonly used in modern residences.

To correlate panel performance with floor covering, the thermal conductance of the floor coverings was needed. A survey of the literature produced no official standard test procedure for determining the thermal properties of textiles, but several methods of using the hot plate apparatus have been proposed.[2,3,4]

Thermal conductances of the floor coverings were determined in a standard guarded hot-plate apparatus[5] using the test procedure outlined by the *American Society for Testing Materials*.[5] The distance between the hot and cold plates was

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[1] Exponent numerals refer to References.

Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

TABLE 1—FLOOR COVERING DATA

CODE	DESCRIPTION	USED IN ROOM	CONDUCTANCE, BTUH PER SQ FT (F)	
			HOT PLATE TEST	MANUFACTURER
COL. (1)	COL. (2)	COL. (3)	COL. (4)	COL. (5)
A. Heavy Carpet (embossed Wilton)	216 Pitch to 27 in. width 8 rows to 1 in. 3 ply wool straight yarn 2 ply wool twist yarn 2 shot; 2 frames total thickness = 0.388 in.		1.28	
B. Light Carpet (velvet twist)	216 pitch to 27 in. width 7 rows to 1 in. 2 ply wool yarn; 2 shot total thickness = 0.300 in.		1.75	
C. Heavy Pad	hair and jute 40 oz per sq yard	C	0.91	
D. Light Pad	hair and jute 32 oz per sq yard		1.27	
E. Rubber Pad	sponge rubber, smooth surface, ¼ in. thick	D	2.52	
F. Heavy Carpet & Heavy Pad		C	0.54 0.53[a]	
G. Heavy Carpet & Light Pad		A	0.64[a]	
H. Heavy Carpet & Rubber Pad		D	0.85[a]	
I. Light Carpet & Heavy Pad			0.60[a]	
J. Light Carpet & Light Pad		C	0.66 0.73[a]	
K. Light Carpet & Rubber Pad		D	1.01 1.03[a]	
L. Asphalt Tile	9 in. × 9 in. × ⅛ in.	B	19.6	36.0
M. Rubber Tile	9 in. × 9 in. × ⅛ in.	A	21.1	36.0

[a] Computed value.

made equal to the measured thickness of the sample as laid on the floor. The thermal conductivities of the asphalt tile, rubber tile, carpets and pads were determined by hot-plate test both individually and in combinations with results as shown in Table 1, Column 4, while the thermal conductivity as supplied by the manufacturer is given in Column 5.

Instrumentation: Approximately 800 copper-constantan thermocouples were used to measure temperatures of room air, heated and unheated room surfaces, attic air, outdoor air, and water in the system. Along the center line of each room perpendicular to the north wall a large number of thermocouples were placed in the ground and under the surface of the concrete slab (see Reference 1, Fig. 3). Along these same center lines, thermocouples were located across the rooms at intervals of 6 in. on the top surface of the floor slabs and at the top surface of the floor coverings at a sufficient number of points to obtain a representative measure of the floor covering surface temperature.

In Rooms A and B thermocouples were installed flush with the top surface of the tile and others flush with the top surface of the concrete floor slab. For the rooms with carpeting, surface thermocouples were pressed slightly into the pile of the carpet. All surface thermocouples were installed so that at least 3 in. of lead wire on each side of the junction was in contact with the surface whose temperature was to be measured.[6]

Forty-four heat flow meters as developed at the ASHAE Research Laboratory[7], made and calibrated at the University of Illinois, were installed both on the top surface of the concrete floor slabs and under the gravel fill. Ten commercial heat-flow meters were installed at the edges of the floor slabs to determine edge losses. Instruments were also provided to measure relative humidity, electrical input to the immersion heaters and control circuits and water-flow rates. Continuous records of the more important temperatures and heat-flow rates were obtained from two recording potentiometers.

OPERATING CONDITIONS

For all tests with bare floor panels, the electric water heaters were adjusted to a heat input rate equal to 1.3 times the estimated design heat loss of the room served. Some increases were necessary in the firing rate for the rooms in which carpets were used. A high limit control was provided in each system and for the tests with bare floor panels it was adjusted to turn off the heater whenever the leaving water attained a temperature of approximately 130 F. Higher water temperatures were required when carpeting was used, and the setting of the limit control was adjusted accordingly. Continuous operation of the circulators was used while electric water heaters were controlled by room thermostats set to maintain a temperature of 72 F at the 30-in. level.

The corridor floor was maintained at the same temperature as the adjoining test room floor by electric cables embedded in the corridor floor. Electrical input was manually controlled by variable transformers. Attic air temperature was maintained at 60 F while the air temperature in the corridor was kept approximately the same as that in the test rooms by convection heaters thermostatically controlled.

Complete thermocouple and meter readings for each room were taken several times each day, and the more important data were recorded over 24-hr periods.

RESULTS

Water Temperatures: Water temperatures were measured in each test room at the inlet and outlet of the coils, and the arithmetic mean of these temperatures was taken to be the average. For each room the average water temperatures for 24-hr test periods have been plotted against the corresponding indoor-outdoor tempera-

ture difference. Curves were fitted by the method of least squares. Fig. 1 shows 2 typical curves representative of those obtained from all test rooms. In all cases it was observed that the greater the thermal resistance of the floor covering, the greater was the increase in the average water temperature for a given decrease in outdoor temperature.

Fig. 2 shows the relationship between the average water temperature at an indoor-outdoor temperature difference of 50 F and the thermal resistance of the floor coverings in Rooms C and D. Also indicated are the standard deviations S_d of the data

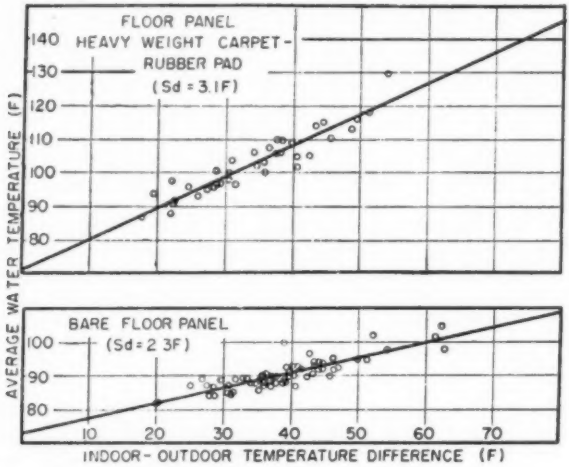


FIG. 1.—AVERAGE WATER TEMPERATURE, ROOM D WITH FLOOR PANEL BARE, AND COVERED

from the line of regression. With the exception of the tests conducted with the ¼-in. rubber pad, where only a relatively small number of test points were obtained, the greater the thermal resistance the larger the standard deviation.

The above-floor heat-output of a floor panel can be expressed in terms of an equivalent thermal transmittance and a temperature difference between water and room air, thus:

$$Q/A = U\,(t_w - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

Q = above-floor heat-flow in Btu per hour.
A = panel area in square feet.
U = equivalent thermal transmittance from water to room air in Btu per hour, per square foot, per Fahrenheit degree.
t_w = average water temperature in Fahrenheit degrees.
t_a = room-air temperature in Fahrenheit degrees.

At 50 F indoor-outdoor temperature difference, the above-floor heat-flow per sq ft of panel area in Room D was 22.8 Btuh. This value was obtained by determining

the average panel surface temperature at 50 F indoor-outdoor temperature difference and the corresponding heat flow as described on pp. 116–17 of the previous paper[1]. The water temperature for tests with a bare floor panel in Room D was 96 F at 50 F indoor-outdoor temperature difference (Fig. 1). Letting R, the equivalent thermal resistance, equal $1/U$ and re-arranging Equation 1:

$$R = (t_w - t_a)/(Q/A) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$R = (96 - 72)/22.8 = 1.05$$

For tests with floor coverings the same equation applies, and R is approximated by the equivalent resistance of the bare floor plus the resistance of the floor covering.

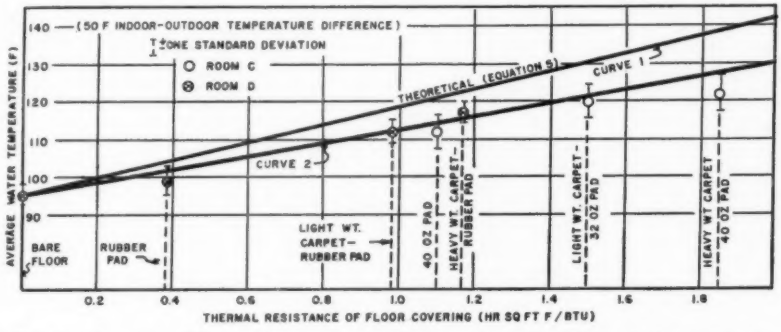


FIG. 2.—AVERAGE WATER TEMPERATURE PLOTTED AGAINST THERMAL RESISTANCE OF FLOOR COVERING IN ROOMS C AND D WITH 50 F INDOOR-OUTDOOR TEMPERATURE DIFFERENCE

T_w becomes the water temperature obtained with the covered panel (t_{wc}). The air temperature and heat flow remain the same. Equation 2 then becomes

$$R + R_{fc} = (t_{wc} - t_a)/(Q/A) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Substituting for R from Equation 2 yields

$$t_{wc} = (Q/A)\, R_{fc} + t_w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

or for Room D

$$t_{wc} = 22.8\, R_{fc} + 96 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Equation 5 is the equation of curve 1 (theoretical) in Fig. 2. Curve 2 has been fitted to the test points.

With the exception of the tests with the bare floor panel, all the test points are below the theoretical line, and the points for Room C are the farthest removed. It is evident that the water temperature required for a material with a given thermal resistance as determined in the guarded hot-plate apparatus was less than that theoretically required. This may be the result of intermittent operation of the heater and nonsteady conditions of testing. Equation 5 is based on the assumption that the concrete slab with a coil can be replaced by a slab with a plane heat source such that one-dimensional heat-flow occurs. Also, differences in the surface condi-

tions for the carpet as tested in the guarded hot plate and as installed in the test room could have resulted in differences in overall conductance.

Thus, it is necessary to know only the equivalent thermal resistance of the bare floor panel and the thermal properties of the floor covering used in order to predict the required water temperature for the covered panel. To make the results more general, the dimensionless ratio $(t_{wc} - t_a)/(t_w - t_a)$ was calculated. This was plotted against the dimensionless ratio of the resistance of the floor covering to the equivalent resistance of the bare panel (Fig. 3).

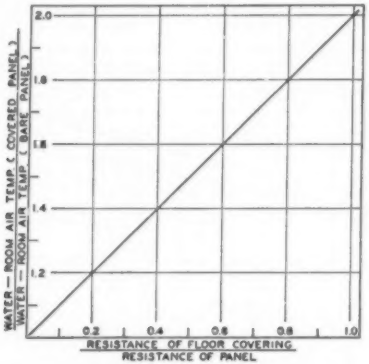


FIG. 3.—EFFECTS OF FLOOR COVERINGS ON WATER TEMPERATURE FOR FLOOR PANELS

The relationship between the water temperatures and resistance of floor coverings (Fig. 2) was obtained at 50 F indoor-outdoor temperature difference. If straight line extrapolation to design conditions (80 F) can be assumed, it would be necessary to take the difference in water temperature for the bare and covered floor under consideration from Fig. 2 and multiply by 8/5 to obtain the water temperature difference at design conditions. The plot of Fig. 3 holds true regardless of the indoor-outdoor temperature difference, because for 2 different panels the ratio of water to room-air temperature differences is equal to the ratio of overall resistance of the panels which is independent of outdoor temperature.

From Fig. 2, curve 2, the average water temperature required with a heavy weight carpet and 40 oz pad at an indoor-outdoor temperature difference of 50 F is 127 F. At 80 F indoor-outdoor temperature difference (design conditions in Urbana, Ill.) the required water temperature would be 8/5 (127 − 72) + 72, or 160 F. Similarly the water temperature at 80 F indoor-outdoor temperature difference would be 110 F for a bare panel. Thus, a 50 F (160 − 110) increase in water temperature would be required when using the heavy weight carpet and 40 oz pad. If this required increase in water temperature were not taken into account in the design of a system, satisfactory performance could not be expected. Re-balancing a system in which one circuit serves a floor panel which has been covered with a floor covering having a high thermal resistance may not be possible by increasing the water temperature in the entire system and throttling down the flow of water to uncovered panels. This

practice would increase the water temperature drop and decrease the effective panel area for those circuits serving bare panels. Hot areas would be apt to appear where the water enters the uncovered panels. This may be undesirable and could be avoided by proper design. It may be necessary to utilize separate zones so that different water temperatures can be maintained in different circuits.

Total Heat Input to Panels: The total energy supplied to the water in the heating system of each test room by the water heater and the circulator was measured by means of watt-hour meters. This energy, minus the heat losses from the insulated heater and connecting piping, was transferred to the panel. Since the losses from the heater and piping were negligible in comparison with the total panel output, the total energy input to the water was assumed equal to the total energy supplied to the panel[1].

Fig. 4 shows total energy supplied to the floor panel in each test room plotted against indoor-outdoor temperature difference. The curves were plotted by the least squares method. Scattering of the test points was due to uncontrollable vari-

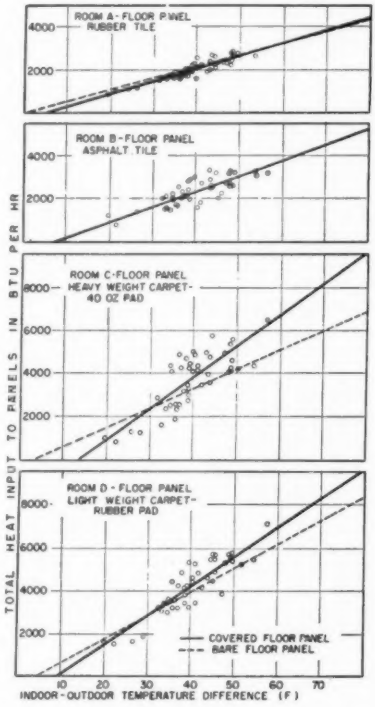


FIG. 4.—TOTAL HEAT INPUT OF FLOOR PANELS AT VARIOUS INDOOR-OUTDOOR TEMPERATURE DIFFERENCES

ations in intensity of solar radiation and wind velocity, and changes in outdoor temperature. Sufficient tests were run so that the fitted curves represent trends for average weather conditions.

Superimposed as dotted lines in Fig. 4 are curves fitted by the method of least squares to data obtained previously for bare panels[1]. In Room A, the rubber tile had practically no effect on the total energy input to the panel and it may be assumed that the effect of the asphalt tile on the total energy input to the panel also was insignificant since the thermal conductivities of asphalt and rubber tile were about the same.

Curves for Room C show that at indoor-outdoor temperature differences greater than about 30 F, the total energy input to the panel was greater for the carpeted floor than for the bare floor, while at indoor-outdoor temperature differences less than about 30 F the reverse was true. The same characteristic was shown in Room D although the difference in energy input was smaller.

The energy input to the floor panel in Room C at design conditions (80 F indoor-outdoor temperature difference) was about 40 percent higher when the panel was covered by the heavy carpet and 40 oz pad than when no floor covering was used. Thus, the insulating effect of the heavy carpet and pad increased the reverse losses

TABLE 2—SEASONAL ENERGY INPUT TO ROOM C (HEAVY WEIGHT CARPET, 40 OZ PAD)

AVG OUTDOOR TEMP, F	AVG NO. OF DAYS PER YEAR	BARE PANEL		COVERED PANEL	
		ENERGY INPUT BTU/HR[a]	ENERGY INPUT BTU/YEAR[b]	ENERGY INPUT BTU/HR[c]	ENERGY INPUT BTU/YEAR[d]
COL. (1)	COL. (2)	COL. (3)	COL. (4)	COL. (5)	COL. (6)
—10 to — 5	0.2	6,580	31,584	9,220	44,256
— 5 to 0	0.4	6,130	58,848	8,500	81,600
0 to 5	0.8	5,700	109,440	7,800	149,760
5 to 10	2.2	5,250	277,200	7,060	372,768
10 to 15	4.6	4,800	529,920	6,330	698,832
15 to 20	7.6	4,370	797,088	5,620	1,025,088
20 to 25	13.6	3,930	1,282,752	4,900	1,599,360
25 to 30	25.4	3,500	2,133,600	4,190	2,554,224
30 to 35	33.8	3,020	2,449,824	3,470	2,814,864
35 to 40	30.0	2,600	1,872,000	2,840	2,044,800
40 to 45	23.4	2,130	1,196,208	2,020	1,134,432
45 to 50	22.6	1,170	927,504	1,310	710,544
50 to 55	20.8	1,270	633,984	590	294,528
55 to 60	19.8	830	394,416	—	—
60 to 65	22.0	390	205,920	—	—
65 to 70	19.0	—	—	—	—
		Total	12,900,238 or 129 therms		13,525,056 or 135 therms

[a] From Fig. 4. [b] Col. 2 × Col. 3 × 24. [c] From Fig. 4. [d] Col. 2 × Col. 5 × 24.
(Based on records of United States Weather Bureau at the University of Illinois. Includes months of January, February, March, April, May, September, October, November, December, from September 1936 to May 1941.)

and heat storage as compared to operation with no floor covering to such an extent that the boiler capacity had to be increased by about 40 percent to satisfy room heat losses at design conditions. The increase in required boiler capacity when the panel was covered by a light weight carpet and ¼-in. rubber pad (Room D) was about 16 percent. No increase in boiler capacity was required for either the asphalt or rubber tile (Rooms A and B).

To make this comparison it was necessary to extrapolate the curves of Fig. 4 about 20 F beyond test points. Past experience has shown that values obtained by these extrapolations are reasonably accurate.

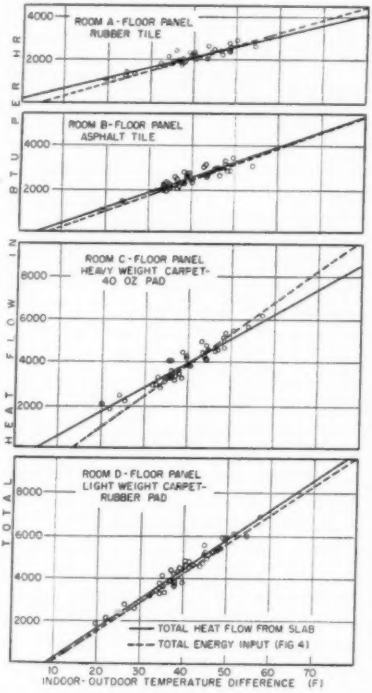

ROOM A - FLOOR PANEL
RUBBER TILE
ROOM B - FLOOR PANEL
ASPHALT TILE
ROOM C - FLOOR PANEL
HEAVY WEIGHT CARPET-
40 OZ PAD
ROOM D - FLOOR PANEL
LIGHT WEIGHT CARPET-
RUBBER PAD
TOTAL HEAT FLOW IN BTU PER HR
TOTAL HEAT FLOW FROM SLAB
TOTAL ENERGY INPUT (FIG 4)
INDOOR-OUTDOOR TEMPERATURE DIFFERENCE (F)

FIG. 5.—TOTAL HEAT FLOW FROM PANELS AS MEASURED BY HEAT FLOW METERS AT VARIOUS INDOOR-OUTDOOR TEMPERATURE DIFFERENCES

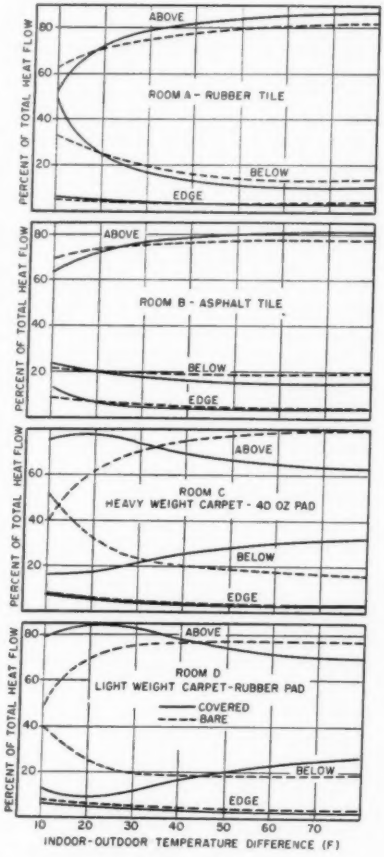


FIG. 6.—PERCENTAGE DISTRIBUTION OF HEAT FLOW FROM BARE AND COVERED PANELS AT VARIOUS INDOOR-OUTDOOR TEMPERATURE DIFFERENCES

If the distribution of outdoor temperatures for a typical winter are known, Fig. 4 may be used to estimate the effect of floor coverings on seasonal fuel consumption. Table 2 shows the estimated energy inputs (fuel consumptions) for Room C when heated by a bare panel and by a panel covered with a heavy carpet and 40 oz. pad. The difference in fuel consumption was about 6 therms per year or an increase of approximately 4.6 percent. This does not include the energy requirement for the initial warm-up period at the beginning of the season. However, unpublished test results have shown that for the most severe warm-up conditions, the energy requirements would not be more than one therm.

Heat Flow From Panels: The total heat flow from the panel as measured by heat meters located on the floor surface, under the gravel fill, and at the exposed edge of the panel were plotted in Fig. 5 against the indoor-outdoor temperature difference. For comparison the total energy input curves of Fig. 4 were reproduced as dotted lines. With the exception of Room C, the total energy input and the total heat flow from the slab were in good agreement. For Room C, at low values of indoor-outdoor temperature difference, the measured heat flow from the panel exceeded the measured energy input, while at high values of indoor-outdoor temperature difference the reverse was true.

Fig. 5 represents results of tests made after the system had been in operation long enough that the initial warm up of the ground had taken place. Also, the majority of the tests were at indoor-outdoor temperature differences ranging from 30 to 50 F with only a few tests at higher or lower indoor-outdoor temperature differences. Obviously, as water temperatures are increased to compensate for the additional heat requirements of the room resulting from a drop in outdoor temperature, an increase in the temperature of the concrete and gravel constituting the floor must result. The higher the thermal resistance of the floor covering, the greater the required change in the average floor temperature per degree change in outdoor temperature. Also, floor coverings having high thermal resistance retard the rate of change of heat flow upward into the room resulting from a change in panel temperature. Both conditions tend to accentuate differences in measured inputs and outputs, and to increase the scatter of test points for Room C.

Distribution of Heat Flow From Panel: Fig. 6 has been constructed from data obtained by totaling the measured heat flow from the floor panel in each room and calculating the percent of the total heat flow in each direction. Superimposed are data for uncovered concrete floor panels[1]. The curves for Rooms A and B show that the percentage distribution of heat flow from the panels covered with either asphalt or rubber tile was practically the same as that for bare panels. But the curves for Room C show that placing a carpet over the panel decreased the percentage useful heat flow from the panel at indoor-outdoor temperature differences of 33 F and greater, and increased the useful heat flow at lower values of indoor-outdoor temperature difference.

The curves of Fig. 6 show comparisons of heat flow from the panel and not energy input to the panel. Because of heat storage in the panel, the two are not the same. The upward heat flow in Room C was approximately the same for both the covered and bare panel at an indoor-outdoor temperature difference of 80 F. This being the case, the increase in total heat flow from the carpeted panel in Room C was

100 × [(Upward Heat Flow/Percent of Total Upward Flow, Covered Panel) — (Upward Heat Flow/Percent of Total Upward Flow, Bare Panel)]/(Upward Heat Flow/Percent of Total Upward Flow, Bare Panel)

Using the values from Fig. 6 this equation becomes

$$100 \times [(1/0.63) - (1/0.79)]/(1/0.79)$$

which is equal to 25.4 percent of the total heat flow from the bare panel at design conditions. Previously, it was shown that carpeting the floor in Room C increased the required energy input at design conditions by 40 percent. The difference between the increase in energy input and the increase in total heat flow from the panel

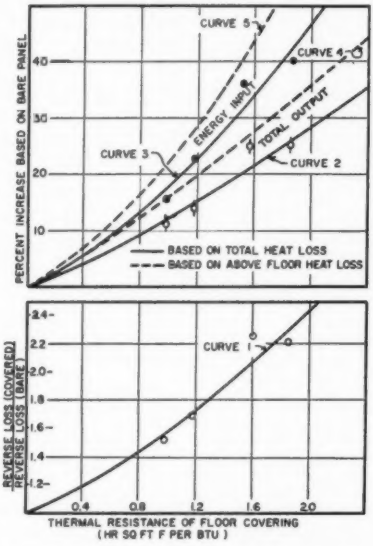


FIG. 7.—EFFECTS OF FLOOR COVERINGS ON HEAT REQUIREMENTS OF FLOOR PANELS

represents increased heat storage in the panel itself. Since heat storage is a function of the change in mean panel temperature which at design conditions was much less for the bare panel than for the covered panel, it follows that heat storage effects were also much smaller for the bare panel than for the covered panel. Increased heat storage for the covered panel was equal to the increase in energy input minus the increase in total heat flow from the panel. The increase in energy input was the difference between the 2 curves of Fig. 4. By assuming that the heat flow from the bare panel was equal to the energy input at design conditions, the increase in total heat flow for the covered panel is given by the difference between the total heat flow from Fig. 5 and the energy input for the bare panel from Fig. 4. By this method of analysis, the increased heat storage for the covered panel in Room C was (9550 — 6800) — (8550 — 6800), or 1000 Btuh. This estimate is conservative because the actual heat flow from the bare panel at design conditions was probably less than the energy input, which would make the increased heat storage somewhat greater than 1000 Btuh.

At design conditions the reverse losses in Room C were 37 percent of the total heat flow from the carpeted panel while for the bare panel these losses were 21 percent of the total. Since the above-floor heat-flow was the same in both cases, the reverse losses for the covered panel were

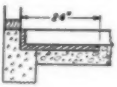
$$[0.37 \times (\text{Upward Heat Flow}/0.63)]/[0.21 \times (\text{Upward Heat Flow}/0.79)]$$

or about 2.2 times as large as those for the bare floor. The increase in reverse losses at design conditions was calculated for each combination of carpet and pad tested. Curve 1 of Fig. 7 shows a plot of the ratio of the reverse losses for the covered panel to the reverse losses for the bare panel *vs.* the thermal resistance of the floor coverings.

Table 3 shows the reverse heat loss for a floor slab utilizing L-type insulation as described in a previous paper[1]. Values are tabulated for unheated and heated slabs with 2 types of floor covering. Heat losses for the heated, covered floor panels were obtained by multiplying the heat loss for the bare panel by the value obtained from curve 1, Fig. 7.

TABLE 3—HEAT LOSSES OF FLOOR SLABS

(Btuh per lineal foot)[a]

FLOOR SLAB	
Bare, unheated[b]	50
Bare, heated[c]	104
Light Weight Carpet, Rubber Pad, Heated	163
Heavy Weight Carpet, 40 oz Pad, Heated	240

[a] 80 F indoor-outdoor temperature difference.
[b] THE GUIDE, 1953, pp. 232–253.
[c] Reference 1.

The percentage increase in total panel output at design conditions resulting from the use of floor coverings is shown by curve 2 of Fig. 7, while the percentage increase in total heat input is shown by curve 3. Since the floor covering did not affect the above-floor heat-loss of the room, the upward heat flow from the panel was unaffected by the covering. Therefore, curve 2, Fig. 7 is a measure of the increase in reverse losses from the panel, and curve 3 is a measure of the increase in reverse losses plus the increased heat storage in the panel. The effect of the floor coverings on heat storage within the panel is represented by the difference between curves 2 and 3.

To apply these results to design, it may be more convenient to express these increases in percent of upward heat flow rather than percent of total heat flow from the panel. To do so requires that the values from curves 2 and 3 be divided by 0.79 (the ratio of upward heat flow to total heat flow for the bare panel). Thus, for a given structure, if the above-floor heat-loss is calculated by the conventional method in THE GUIDE, the increase in heat storage, reverse losses, and energy input may be obtained for any thermal resistance of floor covering from curves 4 and 5.

In general, floor coverings with high thermal resistance had the undesirable char-

acteristic of increasing the reverse losses and heat storage in the panel in cold weather. This resulted in a lower percentage of useful heat to the room, but on a seasonal basis this was compensated at least in part by an increase in the percentage of useful heat obtained during warmer weather. The heat input required to provide for heat storage in the panel is in effect a panel pick-up allowance which should be made when selecting a boiler. Also, if the increase in reverse losses is not accounted for in the heat loss calculations, additional allowance should be made in boiler sizing to compensate for the increased reverse losses.

Application of Results: To summarize the effects of floor coverings on the performance of a floor panel heating system, an illustrative example has been chosen. Three cases are presented: (1) bare concrete slab, (2) concrete slab with carpet and pad having a thermal resistance of 1.0, (3) concrete slab with carpet and pad having a thermal resistance of 2.0. For this example Table 4 shows that use of a

TABLE 4—APPLICATION OF RESULTS

CASE	CONCRETE SLAB ON GROUND		
	COL. (1)	COL. (2)	COL. (3)
Floor Covering	None	Carpet and pad having a resistance of 1.0	Carpet and pad having a resistance of 2.0
Design Temperature Difference, F	80	80	80
Perimeter of House, ft	100	100	100
Design Above Floor Heat Loss, Btuh	40,000	40,000	40,000
Design Average Water Temperature, F (Based on curve 2, Fig. 2)	110	136	165
Reverse Loss at Design Conditions, Btuh (from curve 1, Fig. 7)	$100 \times 104 =$ 10,400	$100 \times 104 \times 1.58 =$ 16,400	$100 \times 104 \times 2.42 =$ 25,100
Total Panel Output at Design, Btuh	50,400	56,400	65,100
Panel Pick-Up Allowance, Btuh (from curves 2 and 3, Fig. 7)	0	$0.054 \times 50,400 =$ 2,720	$0.19 \times 50,400 =$ 9,590
Minimum Net Output of Boiler, Btuh	50,400	59,120	74,690

floor covering having a thermal resistance of 1.0 requires a 26 F increase in water temperature which results in an increase in reverse loss amounting to 16,400 − 10,400 = 6000 Btuh. A floor covering having a thermal resistance of 2.0 requires a 55 F increase in water temperature which results in an increase in reverse loss amounting to 25,100 − 10,400 = 14,700 Btuh. For the floor panel with a carpet and pad having a resistance of 1.0, an additional 2720 Btuh must be provided, while for the floor panel with a carpet and pad having a resistance of 2.0, an additional 9590 Btuh must be provided for heat storage which is a pickup allowance. Therefore, the minimum net output of the boiler for the panels with coverings having resistances of 1.0 and 2.0, respectively, exceeds that for the bare panel by 8720 Btuh and 24,290 Btuh, or about 17 and 48 percent. These figures do not include the normal piping and pick-up factor. Another 30 percent is usually added to provide an allowance for piping and pick-up.

CONCLUSIONS

The following is a summary of the results obtained for the test conditions investigated:

1. Apparent thermal resistance of the bare concrete panel was about 1.05 (F deg) per Btuh (sq ft).
2. Thermal resistance of the combinations of carpeting and pad ranged from 0.40 (F deg) per Btuh (sq ft) for the rubber pad alone to 1.87 (F deg) per Btuh (sq ft) for the heavy carpet and 40 oz jute pad.
3. The thermal resistance of both the asphalt tile and the rubber tile was about 0.05 (F deg) per Btuh (sq ft).
4. Floor coverings, such as asphalt tile or rubber tile, which have a thermal resistance of 0.2 (F deg) per Btuh (sq ft) or less had a negligible effect on the performance of floor panel systems.
5. At design conditions, covering the floor panels with any type of carpeting had pronounced effects on the water temperatures, reverse loss from the panel, and the required boiler size (see Table 4).
6. Covering a floor panel with carpeting did not appreciably increase the seasonal fuel consumption.
7. Because of the large increase in water temperature required when a carpet is applied to floor panels it may be impossible to balance floor panel systems in which carpeting is used in some rooms only, unless the piping is arranged to permit zoning with the use of more than one water temperature.
8. Major effects of carpeting over a bare floor panel on the design and performance of a floor panel system are shown in Table 4.

ACKNOWLEDGMENTS

This report concerns one phase of the panel heating research project which is part of the comprehensive steam and hot water heating investigation being conducted by the *Institute of Boiler and Radiator Manufacturers* in cooperation with the University of Illinois. Eventually, the results of this research project will be compiled in an Engineering Experiment Station Bulletin.

The authors wish to acknowledge especially the assistance of research assistants A. J. Wiegand and Liang-Neng Tao in obtaining and compiling data for this report. Also acknowledgment is made of companies contributing equipment for this study.

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DISCUSSION

L. F. SCHUTRUM, Cleveland, Ohio (WRITTEN): It is always interesting to have the data of one investigator compared with that of another; consequently, the conductances of

floor coverings given in Table 1 of the paper were compared with conductances of similar floor coverings measured in the Society's Environment Laboratory and reported some time ago. The comparisons are good; for example, the heavy pad, 40 oz per sq yard, was found to have a 0.90 conductance compared to 0.91 given in the table. The heavy wool carpet 1.3 vs 1.28. A combination of the carpet and pad 0.53 vs 0.54. The rubber pad and asphalt tile conductance are also in agreement.

The equivalent thermal resistance, R as given by Equation 2 of the paper is a combination of thermal resistance from the water to the slab surface plus a complex film resistance from panel surface to room air. The combination of these 2 resistances determined from experiments in the Laboratory for the same panel output, pipe size and spacing as given in this paper, results in a value of R equal to unity, which is in very good agreement with the value reported for a bare floor of 1.05.

Extrapolated data from field tests made in 1949–51 to measure heat losses from heated floor slabs, indicates that the heat loss of 240 Btu per hr per linear foot, given in Table 3, may be high. Another method of analysing the data reported by the authors leads to a somewhat lower heat loss at design conditions of 80 F indoor-outdoor temperature difference. This method is as follows: The curves shown in Fig. 5 are the total measured heat flow from the slab. Since most of the experimental data shown in the figure were gathered for indoor-outdoor temperature differences of 30 to 50 F, after initial warmup, the curve which fits the points is essentially a performance curve which fluctuates through an average temperature differential of about 40 F. The slope of the curve probably depends upon the rapidity of outdoor temperature variations, thermal storage of the structure above the panel and the earth below, and the rate of heat input to the panel. This is an excellent problem which could be solved by thermal circuit technique. Extrapolation of the heat flow curve (Fig. 5) to design conditions of 80 F differential might have been done by drawing a line from the origin through an average value of heat flow at about 40 F differential.

The heat loss is the product of the total heat flow (Fig. 5) and the percentage of heat loss (Fig. 6). For Room C, heavy weight carpet and 40 oz pad, the loss per linear foot by this method would be about $\frac{(0.37 \times 7800)}{14.5}$ or 200 Btu per hr instead of (2.2×104) or 230 Btu per hr.

W. F. SPIEGEL†, Philadelphia, Pa. (WRITTEN): The studies appear to be quite comprehensive, and the authors' efforts to present the work in such a manner that it can be applied to existing data are most welcome.

There were a few minor points which are not clear to this reader.

Should not the curve 2 in Fig. 7 be labeled *% Increase in Total Output* rather than *Reversal Losses*, of which it is only an indication?

Secondly, the increased heat storage for the covered panel in Room C was computed at 100 Btu per hr on the basis that the heat flow from the bare panels was equal to the energy input at design conditions, while the dotted lines representing the bare panels in Figs. 4 and 5 showed an actual storage surplus of approximately 3,500 Btu per hr for Room C and approximately 1000 Btu per hr for Room D. Although the final figures presented appear absolutely reasonable, it would be interesting to know which approach more nearly describes the actual relative behavior of the 2 types of panels.

H. BUCHBERG*, Los Angeles, Calif.: It might be of practical interest to estimate the amount of additional resistance that must be applied to the slab (under the coils) in order that the useful heat input be the same with floor coverings as it is without. Have estimates of this type been made?

AUTHOR'S CLOSURE (Mr. Sartain): Mr. Schutrum's discussion was both interesting and encouraging. It was enlightening to find that comparisons in thermal conductances

† Charles S. Leopold.
* University of California, at Los Angeles.

of similar floor coverings measured at the Society Laboratory and at the Floor Slab Laboratory were in such good agreement, especially since the 2 techniques employed for this measurement were different. When it was decided to measure thermal conductances by using the guarded hot plate apparatus, the literature was searched for a reference to a standard procedure. Since no such standard was found, the procedure described in this paper was arbitrarily selected.

The excellent agreement between equivalent thermal resistances of similar panels obtained from experiments conducted at the 2 laboratories indicates that performance data obtained under steady state controlled laboratory conditions can be applied to field installations where there may be considerable variation in outdoor temperature and where a non-steady heating condition exists.

Mr. Schutrum's analysis of the data reported here would certainly result in a lower sub-floor heat loss than that reported in Table 3. However, the results of tests performed in the past at both the Research Home and Floor Slab Laboratory have indicated that the heat input performance curves seldom pass through the origin at zero indoor-outdoor temperature difference. This is probably due to the effects of auxiliary heat sources in the structure, solar radiation, and heat storage.

Mr. Spiegel's comment regarding the apparent confusion due to the identification of curve 2 in Fig. 7 is well founded. This curve should have been labeled, *Total Output* rather than *Reverse Losses* for it is truly the percent increase in total panel output due to the increase in reverse losses from the panel as a result of the effects of floor coverings.

The latter part of Mr. Spiegel's comment is not clear since Fig. 4 shows bare and covered panel characteristics while Fig. 5 shows only covered panel characteristics. Heat storage is given by the difference in the 2 curves of Fig. 5 and are in agreement with Fig. 7.

In response to Mr. Buchberg's question regarding the application of additional insulation under the slab, it is assumed that the "useful heat input" to which he referred is actually the percentage of total heat input which is transferred to the test room above the floor. In order for the percent useful heat input to be the same when floor coverings are used as it is with bare panels, the reverse losses from the panels must be equal for the 2 cases.

It was found that at design conditions a mean water temperature of about 160 F was required for the panel with a heavy weight carpet and 40 oz. pad. This was necessary in order to maintain a sufficient temperature difference across the floor covering to transfer the heat required to offset the above-floor room heat loss. An estimate of the amount of sub floor insulation required to produce equal percentage heat distribution with and without floor coverings may be made by assuming that a water temperature of 160 F would be maintained with and without sub-floor insulation. This would not be strictly true because the horizontal temperature distribution along the surface of the slab would be improved with the addition of insulation under the slab. Also differences in heat storage characteristics are neglected in this discussion. With equal water temperatures, the ratio of reverse losses with and without sub-floor insulation would have to be 240/104 (Table 3) in order to obtain equal percent distribution of heat flow. Thus, a ratio of equivalent thermal resistances of $\frac{240}{104} = 2.31$ would be required for downward heat flow.

Since the ground and gravel fill is already a fairly good insulator, it is doubtful that sufficient insulation could practically be applied in order to more than double the equivalent downward resistance to heat flow. In any case, it would still be necessary to increase water temperatures when floor coverings are used, and thus when only a few panels on one zone have floor coverings with low conductances, serious unbalance would result.

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THERMAL DESIGN OF WARM WATER CEILING PANELS

Prepared by A Subcommittee* of the Technical Advisory Committee on Panel Heating & Cooling and Members of the Staff‡ of the ASHAE Research Laboratory

THE DESIGN procedure here presented is based on a solid foundation of research and provides a reliable means for the thermal design of ceiling-type heating panels using warm water as the heating medium. With its simplicity, it retains engineering accuracy appropriate to the usual applications of panel heating in residential and commercial buildings.

The basis of this procedure is a body of experimental data obtained at the ASHAE Research Laboratory in a comprehensive program planned and guided by the Technical Advisory Committee on Panel Heating and Cooling. This work has been reported in the 8 research papers listed in the Bibliography at the end of this paper.

Findings from these studies have been weighed as to importance, and then trimmed to meet the needs of a simple but accurate design procedure.

The simplified procedure provides a panel design to maintain the desired room air temperature for the selected outdoor conditions. Room air temperature is the selected criterion of comfort, and the design procedure is restricted to situations in which the area weighted average temperature of the walls, the floors, and glass does not differ greatly from room air temperature. The room-scale tests, which simulated various conditions of construction and outdoor temperature, showed that this near-equality of the two temperatures normally prevails.

Design Steps

Panel design requires specification of the following: panel area, size, spacing and depth of bury of the pipe or tube, insulation on the rear side of the panel, the tem-

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* *Personnel:* R. L. Maher, chairman; W. P. Chapman, H. T. Gilkey, P. B. Gordon, E. F. Snyder and J. M. van Nieukerken.

‡ *Personnel:* L. F. Schutrum, C. M. Humphreys and G. V. Parmelee.

Presented at the 62nd Annual Meeting of the American Society of Heating and Air-Conditioning Engineers, Cincinnati, January 1956.

perature drop and the mean temperature of the water. The procedure, summarized, is as follows:

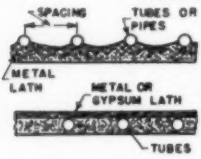
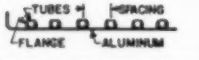
a Calculate the hourly rate of heat loss for each room.
b Determine the available area for the panel in each room.
c Calculate the required unit panel output.
d Determine the required panel surface temperature.
e Select type, size and spacing of pipe or tube, and the insulation for the rear of the panel.
f Determine the required mean water temperature.
g Design the fluid circuit (panel piping and mains) and determine the size of the heating equipment, following conventional practice.

RANGE OF APPLICATION

The design procedure described here is applicable to space heating systems which employ large heated ceiling areas operated at relatively low average surface temperatures. Other limitations are:

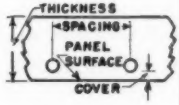
Outside design conditions: Temperatures as low as −30 F.
Room air temperature: 70 to 76 F.
Air changes: No more than two air changes per hour.
Room dimensions: Rooms having normal proportions; ceiling height between 7 and 12 ft.
Room construction: Any type of wall construction and any amount of glass area. (Both however have an effect upon comfort). Conventional interior finishes and furnishings.
Panel construction: Plaster, metal, or concrete panels shown in Tables 1 and 2. Any type of construction above panel. Panel surface exposed to room.

TABLE 1—THERMAL RESISTANCE OF PLASTER AND METAL CEILING PANELS

	Spacing In.	Thermal Resistance to Downward Heat Flow (r_d)					
		Heat Flow Ratio, q_u/q_d					
		0	0.2	0.4	0.6	0.8	1.0[a]
PLASTER PANELS[b] STANDARD GYPSUM PLASTER, THREE COATS	4½	0.30	0.34	0.38	0.42	0.46	0.50
 ⅜ in. (nom.) non-ferrous tube or ½ in. (nom.) ferrous pipe *above* metal lath tied at 8-inch intervals with good tube embedment, or ⅝ in. (nom.) non-ferrous tube *below* metal or gypsum lath.	6	0.45	0.51	0.57	0.63	0.69	0.75
	9	0.75	0.85	0.95	1.05	1.15	1.25
	12	1.15	1.29	1.43	1.57	1.71	1.85
ALUMINUM PANEL[c] ⅛ IN. MINIMUM THICKNESS  pipe or tube installed so that excellent heat conduction to ceiling surface is obtained.	3	0.07	0.07	0.07	0.07	0.07	0.07

[a] See note in table 2.
[b] Recommended maximum inlet water temperature (t_{max}) 140 F.
[c] Recommended maximum inlet water temperature (t_{max}) 220 F.

TABLE 2—THERMAL RESISTANCE OF CONCRETE CEILING PANELS

PANEL CONSTRUCTION 	SPACING IN.	THERMAL RESISTANCE					
		HEAT FLOW RATIO q_u/q_d					
		0		0.5[a]		1.0[a]	
		UP	DOWN	UP	DOWN	UP	DOWN
6 IN. CONCRETE SLAB—1 IN. COVER		r_u	r_d	r_u	r_d	r_u	r_d
½ in. (nom.) non-ferrous tube	9	3.6	0.30	0.9	0.35	0.7	0.45
	12	5.1	0.35	1.1	0.45	0.9	0.55
½ in (nom.), ferrous pipe or ¾ in. (nom.) non-ferrous tube	9	2.6	0.25	0.7	0.30	0.6	0.35
	12	4.0	0.30	0.9	0.40	0.8	0.50
¾ in (nom.) ferrous pipe or 1 in. (nom.) non-ferrous tube	9	2.1	0.20	0.6	0.25	0.6	0.30
	12	3.3	0.30	0.8	0.35	0.7	0.40
	15	4.5	0.35	1.0	0.45	0.8	0.55
1 inch (nom.) ferrous pipe	9	1.6	0.20	0.5	0.25	0.5	0.25
	12	2.6	0.25	0.7	0.30	0.6	0.35
	15	3.6	0.30	0.9	0.40	0.7	0.45
8 IN. CONCRETE SLAB—1 IN. COVER							
½ in. (nom.) non-ferrous tube	9	3.6	0.30	1.0	0.35	0.8	0.40
	12	5.2	0.35	1.2	0.45	1.0	0.55
½ in. (nom.) ferrous pipe or ¾ in. (nom.) non-ferrous tube	9	2.9	0.25	0.9	0.30	0.8	0.35
	12	4.0	0.30	1.1	0.40	0.9	0.45
¾ in. (nom.) ferrous pipe or 1 in. (nom.) non-ferrous tube	9	2.2	0.20	0.8	0.30	0.7	0.30
	12	3.3	0.30	1.0	0.35	0.8	0.40
	15	4.3	0.35	1.1	0.40	0.9	0.50
1 in. (nom.) ferrous pipe	9	1.7	0.20	0.7	0.25	0.7	0.25
	12	2.7	0.25	0.9	0.30	0.8	0.35
	15	3.7	0.30	1.0	0.40	0.9	0.45

[a] Any ceiling panel also acts as a floor panel to the extent of its upward heat flow. If the upward heat flow is high and the space above is occupied, check floor surface temperature for possible foot discomfort (see RESEARCH REPORT No. 1490—Heat Exchanges in a Floor Panel Heated Room, by L. F. Schutrum, G. V. Parmelee and C. M. Humphreys, ASHVE TRANSACTIONS, Vol. 59, 1953, p. 495). Also check effect on heating requirements of the space above. It is not good practice to have the major portion of the upper room's heating requirements supplied by the upward heat flow of a ceiling panel below.

PROCEDURE FOR HOT-WATER PLASTER CEILING PANELS

Step 1. Heat Loss: Calculate the heat loss of each room, following the recommendations contained in the ASHAE GUIDE, but do not include any heat loss through the area covered by the panel.

Example: Three Rooms, A, B, and C, are to have a common water supply temperature; that is, they represent a single zone. They are to be maintained at 72 F air temperature when the outdoor air temperature is zero F. The ceilings of Rooms A and B have floors above them with the space heated to 72 F and an air-to-air U value of 0.25 Btu per (hr) (sq ft) (F). The ceiling of Room C has insulation in the joist spaces and an uninsulated attic space with a combined U value of 0.05 from Room C to outdoor air. Room dimensions and calculated heat losses are as follows:

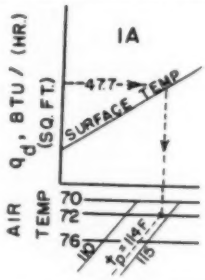
ROOM	FT	BTU/HR
A	11 × 12 × 8	6300
B	11 × 12 × 8	2500
C	15 × 12 × 8	8000

Step 2. Required Panel Output: Divide the heat loss of each room by the maximum ceiling area in the room which can be used as a heating panel. This is the minimum heat output per square foot of panel which will satisfy the requirements of the room. (The panel which requires the highest output per square foot will generally control the design, because the temperature of the water must be high enough to produce the required output from that panel.)

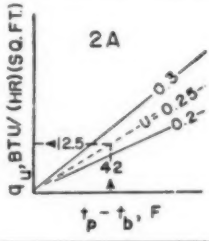
AVAILABLE PANEL AREA		REQUIRED PANEL OUTPUT q_d
ROOM	SQ FT	BTU/(HR) (SQ FT)
A	132	6300/132 = 47.7
B	127	2500/127 = 19.7
C	306	8000/306 = 26.1

Step 3. Panel Surface Temperature: From Fig. 1, find the panel surface temperature needed to yield the required heat output to each room, using the output determined in Step 2 and the design room air temperature. This use of Fig. 1 is shown in illustration 1A.

ROOM	t_p, F DEG
A	114
B	92
C	97



ROOM	U	$t_p - t_b$	q_u
A	0.25	42	12.5[a]
B	0.25	20	6[a]
C	0.05	97	5[a]



[a] In Btu per (hour) (square foot).

Step 4. Upward Heat Flow: Determine the upward heat flow from each panel using Fig. 2. The upward heat flow must be determined in order to obtain the

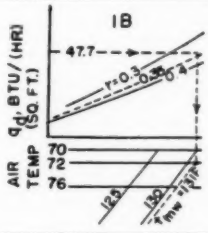
downward panel resistance and to size the boiler. The use of Fig. 2 is given in illustration 2A. Note that the U values to be used with the graph are those for the entire construction of which the panel is a part, and are taken from room air temperature, (t_a) to air temperature above the panel, (t_b'). (See Appendix). The U factors may be obtained from THE GUIDE.

Step 5. Downward Panel Resistance: Assume tentative pipe or tube spacings for the panel construction to be used from those listed in Table 1, choosing the closer spacings when higher outputs are required. In the same table, find the resistance of each panel (r_d), using heat flow ratios calculated from the flow rates determined in Steps 2 and 4, interpolating as required.

ROOM	SPACING IN.	HEAT FLOW RATIO	RESISTANCE (r_d)
A	4½	$\frac{12.5}{47.7} = 0.26$	0.35
B	9	$\frac{6}{19.7} = 0.30$	0.90
C	9	$\frac{5}{26.1} = 0.19$	0.85

Step 6. Mean Water Temperature: For the required panel output (q_d) found in Step 2, the panel resistance (r_d) found in Step 5, and the room air temperature (t_a), find the mean water temperature (t_{mw}) from Fig. 1. This use of Fig. 1 is shown in illustration 1B.

ROOM	t_{mw}, F DEG
A	131
B	110.5
C	119



Step 7. Design Mean Water Temperature: Select a single mean water temperature (t_{dmw}) for each group of rooms which is to comprise a zone, choosing the highest mean water temperature (t_{mw}) of the group subject to the following:

1. If $t_{mw} + \frac{1}{2}(t_i - t_o)$ is equal to or less than t_{max} (see Nomenclature and Table 1), this mean water temperature (t_{mw}) is an acceptable design mean water temperature (t_{dmw}). Proceed to Step 8.

2. If $t_{mw} + \frac{1}{2}(t_i - t_o)$ is greater than t_{max}, go back to Step 5 and select a panel construction which has a lower panel resistance (r_d). This can be accomplished by:

a. Reducing the tube spacing, and/or

b. Decreasing the upward heat flow (q_u) by providing additional insulation above the panel.

If the required mean water temperature is still too high:

a. Reduce the heat loss of the room; and/or
b. Provide supplementary heating.

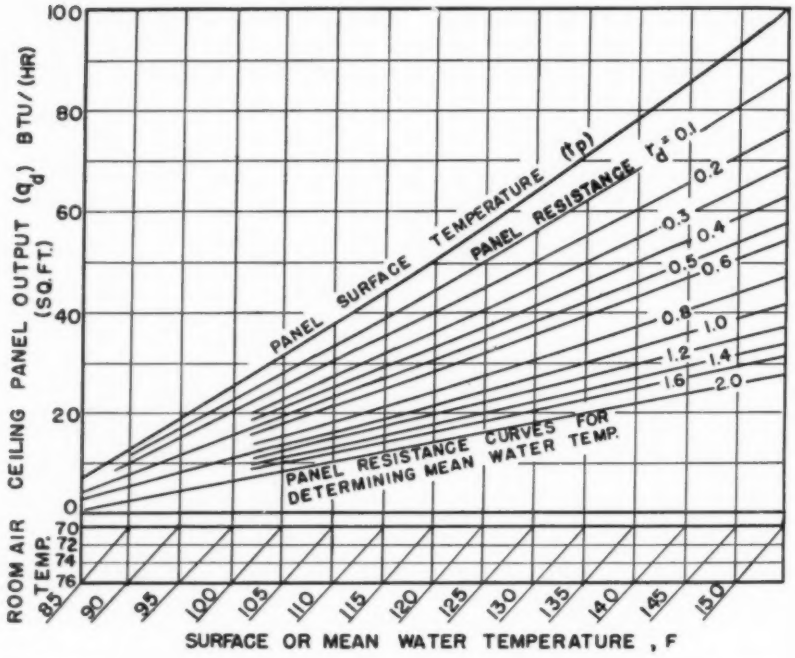


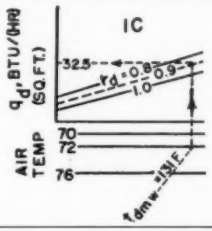
FIG. 1—CEILING PANEL DESIGN GRAPH-PANEL SURFACE TEMPERATURE AND MEAN WATER TEMPERATURE *vs.* OUTPUT DOWNWARD

In the example, assume $t_i - t_o = 15$ F and $t_{max} = 140$ F. Then since $t_{mw} = 131$ F, $t_{mw} + \frac{1}{2}(t_i - t_o) = 138.5$ F and is less than t_{max}. Thus 131 F can be used as the design mean water temperature (t_{dmw}).

Step 8. Design Panel Output: From Fig. 1, find the panel output (q_d) for design mean water temperature (t_{dmw}), room air temperature (t_a) and panel resistance (r_d). Illustration 1C shows how Fig. 1 is used in this step.

The mean water temperature (t_{mw}) of 131 F, found for Room A in Step 6, is used as the design mean water temperature (t_{dmw}). The panel output for Room A which was determined in Step 2 is therefore the design panel output for this room. Design panel outputs for Rooms B and C can be found from Fig. 1 using the design mean water temperature (t_{dmw}).

ROOM	DESIGN PANEL OUTPUT[a] (q_d)
A	47.7
B	32.5
C	33.0



[a] In Btu per (hour) (square foot).

Step 9. Design Panel Area: Divide the room heat loss found in Step 1 by the design panel output found in Step 8.

ROOM	DESIGN PANEL AREA (A_p), SQ FT
A	132[a]
B	77
C	242

[a] Unchanged from Step 2.

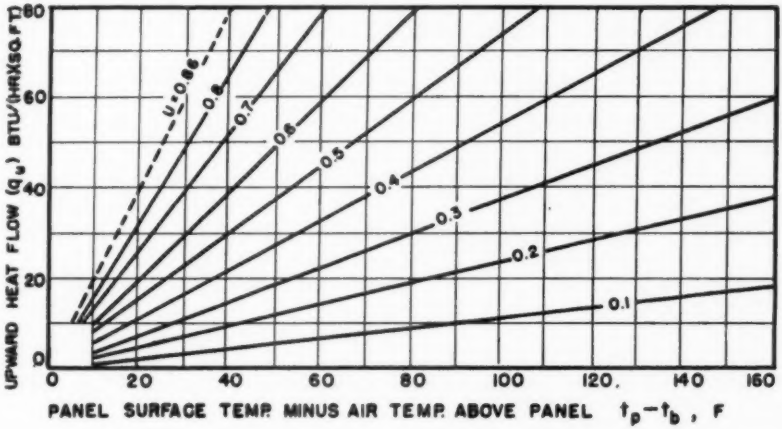


FIG. 2—UPWARD HEAT FLOW FROM PLASTER CEILING PANEL

Step 10. Total Panel Output: Add the heat flow upward (q_u) to the design panel output (q_d) and multiply by the design panel area to obtain the total panel output. If the design panel output is different from the panel output (q_d) used in Steps 3 and 4, the heat flow upward (q_u) should be redetermined.

ROOM	q_u	q_d	$q_u + q_d$	A_p	TOTAL PANEL OUTPUT[b]
A	12.5	47.7	60.2	132	7946
B	9.5[a]	32.5	42.	77	3234
C	5.5[a]	33	38.5	242	9317

[a] Redetermined. [b] In Btu per hour, and equals $A_p(q_u + q_d)$.

Step 11. Fluid Circuit: Design the fluid circuit (panel piping and mains) for a temperature drop of 10 F to 20 F between the water inlet and outlet of the panel (See THE ASHAE GUIDE, Chapter on Hot Water Heating Systems).

Step 12. Boiler Size: Size the boiler according to method outlined in THE GUIDE, chapter on Heating Boilers, Furnaces, Space Heaters. The new Btu rating of the boiler should equal or exceed the total output of all panels plus any other loads on the boiler.

PROCEDURE FOR HOT-WATER METAL CEILING PANELS

Follow the procedure for Hot-Water Plaster Ceiling Panels.

PROCEDURE FOR HOT-WATER CONCRETE CEILING PANELS

The procedure for hot-water plaster ceiling panels cannot be applied in its entirety to concrete ceiling panels because some of the simplifying assumptions regarding the upward heat flow from plaster panels are not valid for concrete panels. The necessary modification of the procedure for plaster ceiling panels is outlined in the following steps:

Step 1. Heat Loss: Follow the procedure for Hot-Water Plaster Panels, Step 1.

Step 2. Required Panel Output: Follow the procedure for Hot-Water Plaster Panels, Step 2.

Step 3. Panel Surface Temperature: Follow the procedure for Hot-Water Plaster Panels, Step 3.

Step 4. Upward Heat Flow Estimate: Determine the coefficient of heat transfer (C) from the upper surface of the slab to air above the panel (temperature, t_b), using factors from THE GUIDE. Estimate the heat flow upward (q_u) by the following equation.

$$q_u = C(t_p - t_b) \text{ Btu per (hr) (sq ft)}$$

Step 5. Upward and Downward Panel Resistance: Follow the procedure for Hot-Water Plaster Panels, Step 5, using Table 2 to find both resistances (r_u and r_d).

Step 6. Mean Water Temperature and Upward Heat Flow: 1. Follow the procedure for Hot-Water Plaster Panels, Step 6; 2. Calculate the heat flow upward from the panel (q_u) as follows:

$$q_u = (t_{mw} - t_b)/(1/C + r_u) \text{ Btu per (hr) (sq ft)}$$

If the calculated heat flow upward differs appreciably from the estimate of the heat flow upward made in Step 4, repeat Steps 5 and 6 using the calculated value.

Step 7. Design Mean Water Temperature: Select the highest mean water temperature (t_{mw}) as the design mean water temperature (t_{dmw}).

Step 8. Design Panel Output: Follow the procedure for Hot-Water Plaster Panels, Step 8.

Step 9. Design Panel Area: Follow the procedure for Hot-Water Plaster Panels, Step 9.

Step 10. Total Panel Output: Follow the procedure for Hot-Water Plaster Panels, Step 10.

Step 11. Fluid Circuit: Follow the procedure for Hot-Water Plaster Panels, Step 11.

Step 12. Boiler Size: Follow the procedure for Hot-Water Plaster Panels, Step 12.

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2. ASHVE RESEARCH REPORT No. 1387—Laboratory Studies on Heat Flow Within a Concrete Panel, by C. M. Humphreys, H. B. Nottage, C. V. Franks, R. G. Huebscher, L. F. Schutrum and D. W. Locklin (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 175).

3. ASHVE RESEARCH REPORT No. 1426—Laboratory Studies of the Thermal Characteristics of Plaster Panels, by C. M. Humphreys, C. V. Franks and L. F. Schutrum (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 363).

4. ASHVE RESEARCH REPORT No. 1491—Further Studies of the Thermal Characteristics of Plaster Panels, by L. F. Schutrum, C. M. Humphreys (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 511).

5. ASHVE RESEARCH REPORT No. 1444—The ASHVE Environment Laboratory, by Cyril Tasker, C. M. Humphreys, G. V. Parmelee and L. F. Schutrum (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 139).

6. ASHVE RESEARCH REPORT No. 1473—Heat Exchanges in a Ceiling Panel Heated Room, by L. F. Schutrum, G. V. Parmelee and C. M. Humphreys (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 197).

7. ASHVE RESEARCH REPORT No. 1499—Effects of Non-Uniformity and Furnishings on Panel Heating Performance, by L. F. Schutrum and C. M. Humphreys (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 121).

8. ASHVE RESEARCH REPORT No. 1516—Effects of Room Size and Non-Uniformity of Panel Temperature on Panel Performance, by L. F. Schutrum and J. D. Vouris (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 455).

APPENDIX

All of the figures and tables included in the Design Procedure are based on data developed at the ASHAE Research Laboratory. The methods by which the figures and tables were derived are described in this Appendix.

Fig. 1. Ceiling Panel Design Graph

The curves of this figure consist of one curve and a family of curves. The single curve shows ceiling panel output (q_d) versus panel surface temperature (t_p). The group of curves relates ceiling panel output to mean water temperature (t_{mw}) for a series of panel

NOMENCLATURE

A_p = panel area, square feet.

C = coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree), from the upper surface of the concrete slab which forms the ceiling panel to point at which t_b is taken. (See definition of t_b).

q_d = panel output (downward heat flow), Btu per (hour) square foot).

q_u = upward heat flow from panel, Btu per (hour) (square foot).

r_d = thermal resistance of panel to downward heat flow (Fahrenheit degrees) (hour) (square foot) per Btu.

r_u = thermal resistance of panel to upward heat flow (Fahrenheit degrees) (hour) (square foot) per Btu.

t_a = design room air temperature, Fahrenheit.

t_b = air temperature, Fahrenheit, at some point above the panel. It may be the temperature of the air in a room or an attic space above the panel, or the outside air temperature. In any case, the value of U or C for the construction above the panel must be consistent with the point at which t_b is taken.

t_i = inlet water temperature, Fahrenheit.

t_o = outlet water temperature, Fahrenheit.

t_{mw} = mean water temperature, Fahrenheit.

t_{max} = maximum water temperature permissible for a given construction, Fahrenheit.

t_{dmw} = design mean water temperature (selected for each zone), Fahrenheit.

t_p = panel surface temperature, Fahrenheit.

U = overall coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree), for the construction between room air and the point at which t_b is taken. (See definition of t_b).

NOMENCLATURE IN APPENDIX

The following terms are used in the Appendix and are not defined in the paper:

Δt = decrease in room air temperature caused by infiltration (see Reference 6, Fig. 9).

$AUST$ = area weighted average temperature of unheated surfaces in room, F.

q_{rd} = radiation component of panel output (downward heat flow), Btu per (hour) (square foot).

q_{cd} = convection component of panel output (downward heat flow), Btu per (hour) (square foot).

q'_{cd} = convection component of panel output (downward heat flow), for no infiltration, Btu per (hour) (square foot).

Δq_{cd} = increase in convection component of panel output (downward heat flow) caused by infiltration, Btu per (hour) (square foot).

f = conventional conductance of panel surface, Btu per (hour) (square foot) (Fahrenheit).

r_p = conventional thermal resistance of plaster in panel (hour) (square foot) (Fahrenheit) per Btu.

a & b = coefficients used in Equation A-8. (see Reference 3, Equation 2).

c = thermal resistance between water in pipe or tube and outer surface of pipe or tube (hour) (square foot) (Fahrenheit) per Btu.

$K, \theta_p, s, m, b_1, L, (q/A)_1, (q/A)_2$ = coefficients used in Equation A-9 (see Reference 1, Equations 3, 4, and 5).

thermal resistances (r_d). The construction at the absicissa serves to adapt the scale drawn for 70 F room air temperature for other design room air temperatures.

The curve of ceiling panel output versus panel surface temperature was plotted from a series of points which were calculated by using the equations which describe panel performance. To locate a point, a room air temperature (t_a) and an average unheated surface temperature ($AUST$) were assumed, and because the latter temperature describes the heating requirements of a room which has the assumed air temperature, the panel surface temperature (t_p) and the panel output (q_d) could be determined by calculation. The procedure followed was:

1. Calculate the panel surface temperature by the equation:

$$t_p = [(t_a + \Delta t) - AUST]/0.08 + AUST. \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-1)}$$

This is an empirical equation derived from laboratory tests and reported in Reference 6, where it appears in Fig. 7. The temperature ($t_a + \Delta t$) is room air temperature for one air change per hour adjusted to room air temperature for no infiltration by adding the decrease in air temperature (Δt) caused by infiltration observed in laboratory tests. The latter observations are presented in Fig. 9 of Reference 6. Infiltration air at 0 F was assumed.

2. Calculate the radiation output of the panel by the equation:

$$q_{rd} = 1.52 \times 10^{-9} [(t_p + 460)^4 - (AUST + 460)^4] \quad . \quad . \quad . \quad . \quad \text{(A-2)}$$

This is the Stefan-Boltzmann equation with a constant which was determined empirically from tests.

3. Calculate the convection output of the panel by the equation:

$$q_{cd} = 0.021 [t_p - (t_a + \Delta t)]^{1.25} + \Delta q_{cd} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-3)}$$

Equation A-3 is an empirical adaptation of the generally accepted equation for convection and based on tests which have been conducted at the Research Laboratory but which have not yet been published. The method of testing the performance of the panels gave the convection output for the condition of no infiltration, which was then increased to the output obtained with infiltration by adding Δq_{cd}. The value of Δq_{cd}, the additional heat flow from the ceiling due to infiltration, is shown in Fig. 4 of Reference 6.

The radiation and convection outputs were added to obtain the total downward output and the latter was plotted against panel surface temperature (t_p) and room air temperature (t_a) to construct the curve.

The following table shows representative results of applying the foregoing equations:

t_a	72	72	72	72
$AUST$	71	72	72.5	73
Δt	3.6	3.6	3.6	3.6
t_p	128.5	117.	111.3	105.5
q_{rd}	61.7	46.9	39.9	32.9
q'_{cd}	3.0	2.2	1.8	1.5
Δq_{cd}	2.2	2.0	1.9	1.9
q_d	66.9	51.1	43.6	36.3

The family of curves relating ceiling panel output (q_d) to mean water temperature (t_{mw}) were plotted from:

$$t_{mw} = t_p + r_d (q_d) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-4)}$$

The construction at the abscissa applies the following corrections:

$$t_p \text{ (at air temperature, } t_a) = t_p \text{ (at air temperature 70 F)} + 5/6\ (t_a - 70\ \text{F}) \quad \text{(A-5)}$$

and

$$t_{mw} \text{ (at air temperature, } t_a) = t_{mw} \text{ (at air temperature 70 F)} + 5/6\ (t_a - 70\ \text{F}) \quad \text{(A-6)}$$

This linear correction is sufficiently accurate for practical purposes in the range of air temperatures 70 F to 76 F.

In the ASHAE Environment Laboratory the test room conditions were (Reference 5):

1. Dimensions 24 ft, 6 in. × 12 ft × 8 ft;
2. Emissivity of room surfaces 0.92‡;
3. Entire ceiling a heating panel;
4. All unheated surfaces at a uniform temperature *i.e.* no colder areas simulated windows or doors;
5. No furnishings;
6. Infiltrating air 0 F;
7. Stable (steady state).

Tests established that the effect on panel output was negligible if room dimensions were changed, if only part of the ceiling were heated, if unheated surfaces were not at uniform temperatures, and if room furnishings were added (References 7 & 8). The emissivity used is representative of the emissivities of conventional wall surfaces and furnishings and is applicable to all but those which have special heat reflective properties. The effect of infiltration of air at temperatures other than 0 F and the effect of different rates of infiltration can be estimated from data reported in Reference 6.

Fig. 2. Upward Heat Flow from Plaster Ceiling Panels

The upward heat flow from a plaster ceiling panel is:

$$q_u = \frac{t_p - t_b}{\frac{1}{U} - \frac{1}{f} - r_p} \quad \text{(A-7)}$$

The average temperature of the surface of the upperside of the panel was found to be practically the same as panel surface temperature (t_p) and for that reason the latter is used. For convenience, the thermal resistance between the upperside of the panel and the air above the panel is expressed in terms of the conventional thermal resistance from room air to air above the panel (U), minus the resistance of the air film at the panel surface ($f = 1.95$), and also minus the resistance of the plaster of the panel ($r_p = 0.14$). These latter 2 resistances are constant for the plaster panels shown in Table 1 and for that reason the graph can be constructed in terms of q_u, U and $(t_p - t_b)$.

The actual upward heat flow from metal ceiling panels will be slightly less than that shown on the graph because r_p for metal panels is very small.

Table 1. Thermal Resistance of Plaster and Metal Ceiling Panels

The thermal resistance of plaster panels to downward heat flow (r_d) was calculated from:

$$r_d = a + (q_u/q_d)\ b + c \quad \text{(A-8)}$$

This is essentially Equation 2 of Reference 3 with term c added. The latter is a small resistance added to adjust for the resistance to heat flow from the water in the tube to

‡ RESEARCH REPORT NO. 1528—Measurement of Angular Emissivity by Aydin Umur, G. V. Parmelee and L. F. Schutrum ASHAE TRANSACTIONS, Vol. 61, 1955, p. 111).

the outer surface of the tube which was not measured in the laboratory (see THE ASHAE GUIDE Chapter on Heat Transfer). The values of a and b, determined by taking the average of the results obtained in the tests of the three better panels described in Reference 4, are:

SPACING IN.	a	b
4½	0.280	0.175
6	0.415	0.295
9	0.720	0.515
12	1.125	0.695

The thermal resistance of the metal panel to downward heat flow was determined from routine tests of the heat transfer through the panels of the ASHAE Environment Laboratory, for which no published paper is available.

Table 2. Thermal Resistance of Concrete Ceiling Panels

The thermal resistance of concrete panels to downward and upward heat flow (r_d and r_u) were calculated from:

$$r_d \text{ or } r_u = (1/k)[(k\theta_p s/m) + (b_1^2/L)] + (1/k)[(q/A)_2/(q/A)_1]\,[(k\theta_p s/m) + (b_1^2/L) - b_1] + c \quad \text{(A-9)}$$

This equation was derived from Equations 3, 4 and 5 of Reference 1. It will yield the resistance of the panel to either downward or upward heat flow depending upon which surface of the panel is identified with subscript (1). The value of $k\theta_p s/m$ was determined from the graphs, Figs. 6a to 6f of Reference 1. The thermal resistance between the water in the tube and the surface of the tube is the term c in Equation A-9 which is the same as term c in Equation A-8.

DISCUSSION

S. C. HUDDLESTON,* New Kensington, Pa., (WRITTEN): The authors have developed, from a mass of experimental data, a simple procedure which should be very useful in the design of ceiling panel heating systems. The only data which they present for aluminum panels are for a panel thickness of 1/16 in. and a tube spacing of 3 in. These were obtained from the performance of the enclosed panels of ASHAE Environment Laboratory, which were installed for technical control rather than economical comfort control. Actually there are many practical aluminum ceiling panel systems operating in which the aluminum is much thinner and tube spacings are as large as 24 in.

In a previous ASHAE Paper (No. 1425), "Aluminum Ceiling Panels for Heating and Cooling," the discusser and his associates presented data and an analytical approach which makes possible the presentation of additional aluminum ceiling panel information in the same form as used by the authors for plaster and concrete panels and the single aluminum panel. For metal panels in which there is no heat flow upwards, it can be shown by combining the authors' Equation A-4 with Equations (4), (5), (6) and (6a) of Paper No. 1425 that:

$$r_d = \frac{1 - R}{hR} + \frac{s}{c} \quad \text{(D-1)}$$

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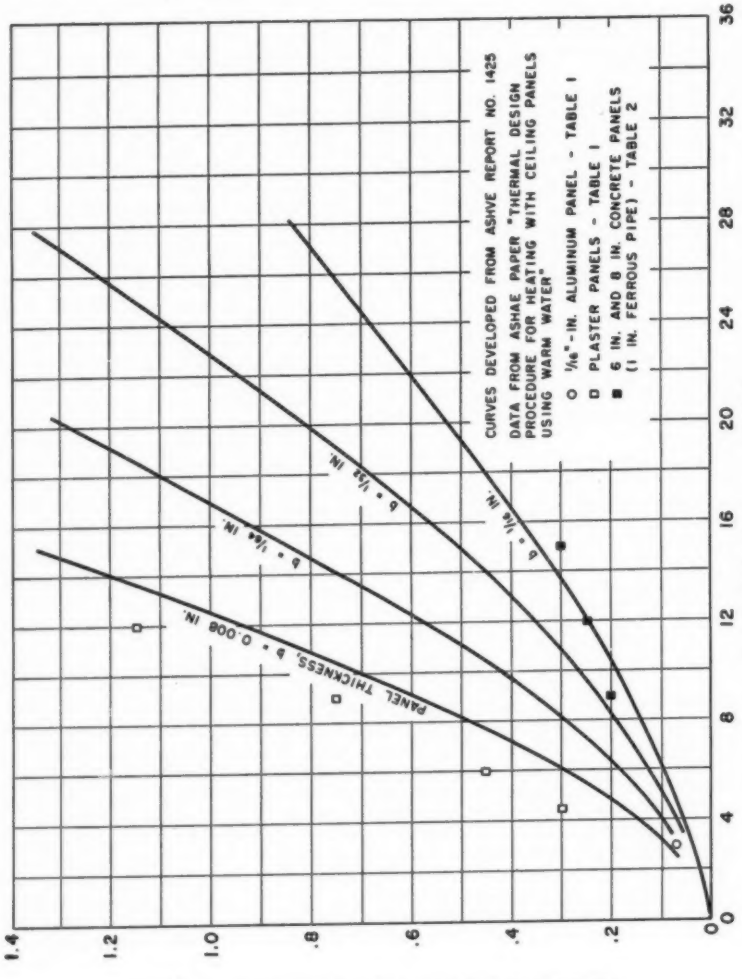

FIG. A—EFFECT OF PANEL THICKNESS AND TUBE SPACING ON THERMAL RESISTANCE, r_d, OF ALUMINUM PANELS FOR $q_u/q_a = 0$

in which

r_d = thermal resistance of panel to downward heat flow (Fahrenheit degree) (hour) (square foot) per Btu.
s = tube spacing, feet.
c = heat transfer coefficient for flow of heat from the water in the tube through the water film and metallic joint to the panel surface directly under the tube, Btu per (hour) (foot of tube) (Fahrenheit degree).
h = heat transfer coefficient from the surface of the panel to the unheated room surfaces at an average temperature, AUST, Btu per (hour) (square foot) (Fahrenheit degree).
k = thermal conductivity of aluminum, Btu per (hour) (square foot) (Fahrenheit degree per foot).
b = panel thickness, feet.

$$R = \frac{\tanh \frac{s^2h}{4kb}}{\frac{s^2h}{4kb}}$$

The curves of Fig. A were calculated from Equation D-1 to show the variation of r_d with tube spacing, s, and panel thickness, b. The thermal conductivity of aluminum, k, was taken as 130 Btu per (hr) (sq ft) (deg F per ft) and an average value of $h = 1.1$ Btu per (hr) (sq ft) (deg F) was calculated from the top curve of the authors' Fig. 1. A value of $c = 8.0$ Btu/(hr) (deg F) (ft of tube length) was selected from unpublished work of the discusser to be representative of the conductance of a good mechanical joint between tube and panel face.

These calculations result in a value of 0.04 for r_d under conditions of $s = 3$ in. and $b = 1/16$ in. as compared to a value of 0.07 shown in the authors' Table 1. This reasonable agreement is indicated in Fig. A by the proximity of the plotted open circle and the curve for $b = 1/16$ in.

The open squares in Fig. A were taken from the authors' Table 1 for plaster panels. The solid squares were taken from the authors' Table 2 and represent the best performance presented for concrete panels. It is interesting to note that the thermal resistance of the plaster panels is equivalent to that of an aluminum panel with a thickness somewhat less than 0.008 in. The thermal resistance of a 1/16 in. aluminum panel is approximately equivalent to the thermal resistance of 6- and 8-in. concrete panels with 1 in. (nom.) ferrous pipe.

The data presented for aluminum panels in Fig. A should permit the application of these design procedures to aluminum panels of different thicknesses and having different tube spacings than the single condition included by the authors.

P. B. GORDON, New York, N. Y.: I would like to thank the hardworking members of the Subcommittee and the members of the Laboratory staff for their excellent work in preparing this design procedure. Though my name is listed as a member of the Subcommittee, I was not one of the hardworking members of the Committee. My present comments are not intended as those of a member of the Subcommittee.

It is of interest to note that a parallel paper on design procedure is being prepared by this Subcommittee and the Laboratory's staff with regard to the performance of concrete warm water floor panels. The format of this new design procedure very closely parallels the procedure presented here for ceiling panels. A third paper is planned for a further design procedure on the performance of panel cooling.

This leads me to my next point. This mechanism of presenting papers is necessary to permit transferring research and design data into a format that permits ultimate use in THE GUIDE. While it is time-consuming, it is necessary that these channels be followed. Presenting these design papers in this manner exposes all the problems to everyone's views, that is, they can be criticized or amplified as was done so ably by Mr. Huddleston.

Though the Subcommittee started with the idea that they would like to have a very

simple approach, the ultimate development indicated that the procedure cannot be streamlined too far. Yet, the information is all there and, for those who so desire, parts of the approach can be eliminated with further simplification.

For example, the University of Illinois' work was based on a standardized panel arrangement and construction, that is, floor panels using ¾ in. pipes on 12 in. centers, and ceiling panels using ¾ in. pipes on 6 in. centers. With this fixed arrangement of panels and a fixed inlet water temperature, you eliminate many of the steps that were necessary in the more complete design procedure submitted here.

Yet, in order to provide for an all-inclusive arrangement which will permit anyone later on to juggle the values in any way he would like, it becomes necessary to include much of the information presented in this paper.

D. L. LINDSAY, Montreal, Que., Canada: I have just been comparing some of the data in this paper with similar data which we have been using in possibly over 100 installations of ceiling panels largely, as Mr. Gordon has mentioned, standardizing on a given water temperature, a given tube spacing, and a given size of tube; and I find that it is very much in agreement.

I do find that we have from time to time been faced with the same problem of meeting an unusual condition and having to spend an awful lot of time.

I find that the information as presented here will be very easy for the design engineer to follow. I do encourage the Society to carry on and provide further data on such aluminum panels, perforated aluminum panels, wherein acoustic properties are desired and other types of construction where floor and ceiling conditions are to be considered on multi-story buildings.

We have had many cases of barracks buildings where we have tried to provide upward and downward flow in the right proportion, and we have found that insulating the floor panels in order to proportion the upward flow to the point where the floor surface temperatures were not uncomfortable, and the downward flow to be adequate to handle the heat loads, has been a very satisfactory means, and I would like to see (possibly we can provide the information which we have used) that the information which we have used could be confirmed by a Subcommittee and the members of the Research Laboratory staff.

E. L. SARTAIN, Racine, Wisc.: I would like to make one point regarding this design procedure. Studies made at the Society Laboratory indicated that with poor tube embedment in ceiling panels having tubes located above the lath comparatively poor performance was obtained. This conclusion was also verified at the University of Illinois.

Since no difference in performance is indicated in this paper with tubes located above or below the lath, it should be stressed that when tubes are located *above* the lath, good tube embedment *must* be obtained.

E. R. KAISER, Cleveland, Ohio: There may be in the minds of some of you a question about whether the information in the paper can be applied to electric-panel heating. I think Mr. Schutrum has given thought to that and perhaps Mr. Maher; so either one could answer it very well, I believe.

H. BUCHBERG,† Los Angeles, Calif.: Would one of the authors please comment on the importance of including radiation exchange with an occupant as well as the air temperature as a criterion for design?

F. J. LINSENMEYER, Detroit, Mich.: In connection with Table 1, we have a resistance to heat flow which is made up of surface resistance and material resistance; i.e. between water and tube and plaster and surface, and then in the case of the aluminum panel we have what would, I imagine, represent mostly resistance out of the tube into the panel,

† Department of Engineering, University of California at Los Angeles.

and then the remainder being what would be represented by an emissivity factor of the aluminum surface.

I don't find from a cursory examination that the aluminum is painted or has any surface treatment. I presume the back would be mill finished, but is the face also polished or mill finished? Is the resistance 0.07 made up mainly of the aluminum, the upward face of the aluminum, or the emissivity of the lower side of the aluminum?

I don't quite see how we can jump from resistance from materials, plaster and surface, over to what appears to be just the surface.

I have another comment: Even though we are not prepared to discuss comfort at this time, I do notice that in the case of ambient room temperature you are dealing with temperatures from 70 to 76. Can we interpret that as a common temperature for design today as against some of our older thoughts that ambient temperatures shall be much lower than, say, other forms of heating?

AUTHORS' CLOSURE (R. L. Maher): In regard to Mr. Huddleston's discussion the authors would like to thank him very much for his addition to the paper. I might point out that Tables 1 and 2 were designed especially for extension. That is, when we started out we hoped that the framework that was devised would later be amplified in THE GUIDE to take care not only of aluminum panels but any other type of panel that might be developed in the future.

I think I should mention that when I say *we*, I mean all of the authors of this paper. That includes Mr. Schutrum, Messrs. C. M. Humphreys and G. V. Parmelee of the Laboratory, and the members of the Committee: myself, Mr. W. P. Chapman, Mr. H. T. Gilkey, Mr. P. B. Gordon, Mr. E. F. Snyder, and Mr. J. M. van Nieukerken. All of these people contributed to this paper, and all of them should share in any compliment or even brickbats that may result.

Mr. Sartain's discussion is certainly a valuable addition to the paper. When we were starting to write the paper, we were faced with a multitude of items which might be included; such points as the philosophy of design, comfort, the matter of embedment, and so on. . . . We chose to stick as close as possible to the single purpose of presenting a design procedure. We hope and we trust that when the paper is adapted to THE GUIDE the supplementary information concerning embedment of the tube, the matter of plaster mixes, more detail on maximum temperatures for gypsum plaster, and similar subjects will be included. We felt that to obtain our objective at this time we should keep close to one subject; namely, the thermal design, not the construction.

In answer to Mr. Kaiser's discussion, I believe you will find that if you stop to analyze the paper carefully—and I don't mean to imply that Mr. Kaiser has not done so, but looking at the details, looking behind the curves a little bit, if you will, for data applicable to electric heating, you will find it.

You have to put so many Btu into the room, even if you measure them in electrical terms. From the required input, you can get the panel surface temperature that will be generated. This will enable you to check the temperature of the panel, assuming that it is running at steady state. All it will require is a little bit of manipulation.

Replying to Professor Buchberg's discussion I think we will by-pass it for the moment. Rather than discuss comfort, I think we should just say that this is something that will be taken up by the Committee charged with sensations of comfort in due time.

As to the point raised concerning the nature of the aluminum surface used the detailed answer can be found in the paper which describes the construction of the ASHAE Environment Laboratory. The interior finish of the Laboratory, that is the aluminum paneling, has normal reflectivity. That is, what you would expect to find in conventional wall coverings and ceiling coverings in residences. Thus so far as Table 1 is concerned, the panel is made of aluminum but is finished to have the normal emissivity of the conventional interior finishes found in residences.

Concerning the temperatures used in the procedure the 70 to 76 F is what we believe to be the conventional operating range for inside residential comfort with radiant heating systems. We all know when radiant heating systems first came out, we were talking

about comfort at air temperatures of 65 F. However, experience has shown that you usually cannot get wide separation of air temperature and the mean radiant temperature in the room without copious mechanical ventilation; so that the fiction of lower air temperatures has been fairly well dispelled, that is, in terms of 5 degrees or more; accordingly, we feel the temperature range 70 to 76 F is more correct. A more correct temperature range of course will be determined in the course of the studies of comfort.

1560

No. 1560

HEAT GAIN THROUGH GLASS SKYLIGHT FENESTRATIONS

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, INC., at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio.

By D. J. VILD* AND G. V. PARMELEE**, CLEVELAND, OHIO

INCREASING USE of skylights in industrial and school buildings, and to some extent in residences, has developed a need for data on heat gain through this type of fenestration. A wide variety of fenestrations are in use, including sheet glass, sheet glass combined with some form of light control device, glass block, reinforced plastic sheet (usually corrugated), and plastic domes of various types. Skylights are particularly suited to large-area single story buildings in that they provide natural illumination to interior zones which cannot be reached by light from sidewall fenestrations. Early in 1954 a comprehensive program was formulated by the Technical Advisory Committee on Heat Flow through Glass□ to develop the heat flow data needed by air-conditioning design engineers and architects. This program constitutes an extension of previous investigations which have been carried on at the Research Laboratory.

Proper skylighting requires an optimum combination of heat exclusion, light transmittance, and light distribution. This condition should exist in a room during as much of the period of occupancy as possible. It is desirable that the light entering the enclosure be evenly distributed and at a practical intensity level.

The results reported here are for 4 types of light-controlling skylights and 3 sheet-glass skylights.

TEST APPARATUS

The equipment used was the ASHAE solar calorimeter, previously described[1] in detail. Basically the calorimeter is as shown in Figure 1. The calorimeter sur-

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** Senior Research Supervisor, ASHAE Research Laboratory, Member of ASHAE.
□ Personnel: R. A. Miller, *chairman;* N. B. Hutcheon, *vice chairman:* W. J. Arner, E. W. Conover, R. B. Crepps, D. D'Eustachio, W. B. Ewing, J. E. Frazier, J. S. Herbert, R. W. McKinley, E. C. Miles, F. W. Mowrey, J. P. Stewart, H. B. Vincent, O. F. Wenzler, C. J. Youngblood, Jr.
Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.
[1] Exponent numerals refer to References.

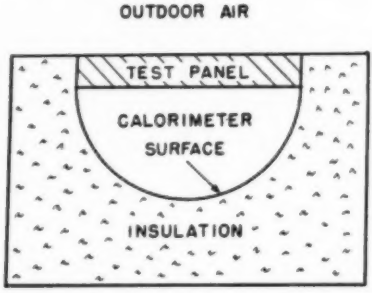


FIG. 1—SCHEMATIC DIAGRAM OF SOLAR CALORIMETER

face is made of sheet aluminum to which are welded aluminum tubes. A glycol-water solution is circulated through these tubes and, by means of an electronic temperature controller, is maintained at a constant inlet temperature, in most cases about 75 F. The total heat gain was found from the flow rate and the difference between entering and leaving temperatures of the coolant. Corrections for heat leakage through the back of the calorimeter and the edges of the fenestration were found by direct measurements and calibration.

It is possible to rotate and tilt the calorimeter to simulate any desired position of the sun. This makes it unnecessary to wait until a particular time of day to obtain a required solar altitude and azimuth.

WEATHER CONDITIONS

Weather conditions affecting the thermal behavior of the skylight were recorded. Diffuse and total solar radiation were measured by 2 multi-junction Eppley pyrheliometers mounted adjacent to the skylight. At regular intervals they were checked against each other. Low-temperature radiation was obtained by means of a convection-compensated radiometer developed at the Laboratory. Air temperature was measured by both an aspirated thermocouple and an aspirated psychrometer. The latter also yielded the air wet-bulb temperature. Wind velocity was determined with an integrating cup-type anemometer and wind direction by a wind vane. Cloudiness and other sky conditions were noted.

Outputs of thermocouples and pyrheliometers were read at regular intervals on an electronic self-balancing potentiometer. In addition, values most subject to fluctuation, such as the solar radiation and the temperature differential between the inlet and outlet coolant, were recorded on a high-speed recording potentiometer and were averaged.

TEST PANELS

During this program tests were conducted on 4 light-controlling fenestrations designed for use as skylights. Each panel measured approximately 4 ft sq and all but the diffusing-type block panel had the characteristic of requiring a particular orientation for proper performance. Hence, each of the panels, with the one ex-

ception had an end designated as the *south* end. Brief descriptions of these skylights follow.

The thermal performance of sheet glass used as skylight fenestrations was sudied experimentally in earlier research, and the test setup and data on solar energy transmittance have been previously reported[2].

Prism-Type Glass-Block Panel: This panel consisted of 16 blocks measuring $10\frac{5}{8} \times 10\frac{5}{8} \times 3\frac{1}{16}$-in. It was prefabricated with the blocks supported on 12-in. centers by a grid of aluminum T-sections. An insulating cement was used as a filler between the blocks. The surface area of the panel was 78 percent glass block, the remainder being either aluminum or sealing mortar which covered the cement. Light control was by means of light-directing prisms on the lower side of the upper face and a light-diffusing fiber glass cavity divider (see Fig. 2).

Diffusing-Type Glass-Block Panel: Nine $11\frac{3}{4} \times 11\frac{3}{4} \times 3\frac{7}{8}$-in. glass blocks supported by reinforced sand-and-gravel concrete made up this panel. The blocks were on 15-in. centers with $3\frac{1}{4}$ in. between blocks, making 54 percent of the surface glass. This panel was typical of those in use with this type of glass block. In other installations the distance between blocks would range from $2\frac{1}{4}$ to 6 in. For this reason, resuts are presented for the glass block alone as well as for the panel, with a method included to account for various widths of concrete. Light control for this block was by means of a ribbed inner surface and a green-tinted fiber glass cavity divider (see Fig. 2).

Compound Louvered Device with Regular Plate Glass (*Glass No.* 1): This panel was a saw-tooth arrangement of sheet aluminum panels and aluminum sun-screens over a sheet of regular plate glass. A description is contained in Fig. 2. Both surfaces of the opaque panels were anodized to obtain diffusing surfaces. The sun-screen surfaces were specular and treated to resist oxidation. These panels are prefabricated in different lengths and widths, but with the same cross-sectional configuration of each saw-tooth. Light control was accomplished by the light-diffusing properties of the sheet aluminum panels and by the shading provided by the sun-screen.

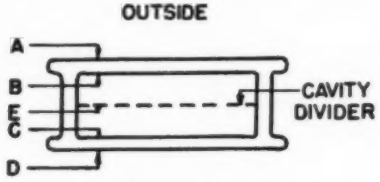
Compound Louvered Device with Hammered and Etched Flint Glass (*Glass No.* 21): Same as just described, but with a different type of glass.

TEST PANEL TEMPERATURE MEASUREMENTS

Glass-block temperatures were measured by thermocouples located on blocks on a diagonal of the panel. The top and bottom surfaces of these blocks each had 4 thermocouples cemented into shallow grooves cut into the surfaces and placed to measure temperatures at centers of equal areas. These thermocouples could be read individually or in parallel for each surface to yield an average glass-surface temperature.

The prism-type block panel had 8 thermocouples cemented to the aluminum grid on each surface, spaced to read temperatures of equal areas. On the top surface 4 thermocouples were also cemented to the sealing mortar between the grid and the block; the bottom surface had no seal showing between the aluminum grid and the glass block. Thermocouples could be read individually or in parallel for each surface and material.

The concrete portion of the diffusing-type glass-block panel had 6 thermocouples cemented to each surface. Groups of 3 were placed at $\frac{1}{6}$, $\frac{1}{2}$, and $\frac{5}{6}$ of the distances between blocks. These yielded temperatures at centers of equal spaces and



Glass Block Patterns

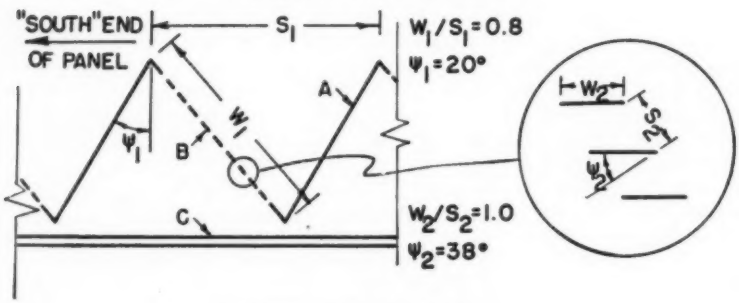
Prism Type

A. Smooth surface
B. Light-directing prisms
C. Light-diffusing surface
D. Light-diffusing surface
E. Clear fiber glass screen

Diffusing Type

A. Smooth surface
B. Stippled surface
C. Stippled surface
D. Narrow-ribbed surface
E. Green-tinted fiber glass screen

Compound Louvered Device



A. Diffuse reflecting aluminum
B. Specular aluminum sunscreen
C. Flat glass sheet

Flat Glass Sheets

Glass No.	*Description*
1	Regular plate glass
11	Hammered heat absorbing glass
12	Hammered and etched heat absorbing glass
21	Hammered and etched flint glass

FIG. 2—DESCRIPTION OF GLASS BLOCKS, COMPOUND LOUVERED DEVICE, AND SHEET GLASSES

when averaged gave an average temperature of the concrete surface. Thermocouples for each surface could be read individually as well as in parallel.

The compound louvered device had 4 thermocouples on each of 2 sun-screen panels and 2 solid panels. There were also 2 thermocouples on each of the 2 side panels. Couples on each pair of similar panels could be read individually or in parallel. Again the couples were located to give temperatures of equal areas. The glass used in conjunction with this device had 4 thermocouples cemented in shallow grooves on each surface at centers of equal areas. The thermocouples for each surface could be read individually or in parallel.

RESULTS

Results are presented to enable predictions of the total instantaneous heat gain through the 4 previously-described light-controlling skylights and 3 sheet-glass skylights. The gain is divided into 2 parts; the transmitted solar energy (direct and diffuse) and the convected and reradiated gain due to the combined effects of absorbed solar radiation, outdoor air temperature, low-temperature radiation exchange with the environment, and wind velocity.

TABLE 1—DIFFUSE TRANSMITTANCE VALUES OF LIGHT-CONTROLLING AND SHEET-GLASS SKYLIGHTS

TYPE OF FENESTRATION	RANGE OF VALUES	AVG.
Prism-Type Glass Block Panel	0.11–0.24	0.17
Diffusing-Type Glass Block Panel	0.07–0.10	0.09
Louvered Device & Glass #1	0.10–0.11	0.11
Louvered Device & Glass #21		0.08[a]
Glass #1	0.65–0.68	0.67
Glass #12	0.13–0.24	0.18
Glass #21	0.47–0.59	0.48

[a] Calculated assuming diffuse transmittance of louvered device-glass combinations varies directly as diffuse transmittance of glass sheets alone.

Diffuse Solar Transmittance: An average diffuse solar transmittance was determined for each of the light-controlling panels from a number of tests run on overcast days. The transmitted diffuse solar radiation was obtained by subtracting the convected and re-radiated gain to the calorimeter surface from the total heat gained by the calorimeter, corrected for edge and back losses. Diffuse solar transmittances for the fenestrations tested in this program and for the sheet-glass fenestrations previously reported[1, 2] are given in Table 1.

Direct Solar Transmittance: Because of the directional properties of the prism-type glass block and the compound louvered device, it was necessary to determine the direct solar transmittances for all combinations of solar altitude and azimuth. Solar altitude, solar azimuth and a third term, profile angle[3], are shown in Fig. 3. The solar azimuth is measured from the *south* end of the panel, either clockwise or counter-clockwise with values ranging from 0 to 180 deg. Solar altitude is the angle of the sun above the horizon, and values range from 0 to 90 deg, being 90 deg at normal incidence. Profile angle, best defined by Fig. 3, ranges from 0 to +90 deg when the sun is south of east-west axis of panel and from 0 to −90 deg when the sun is north of east-west axis (*i.e.* solar azimuth greater than 90 deg).

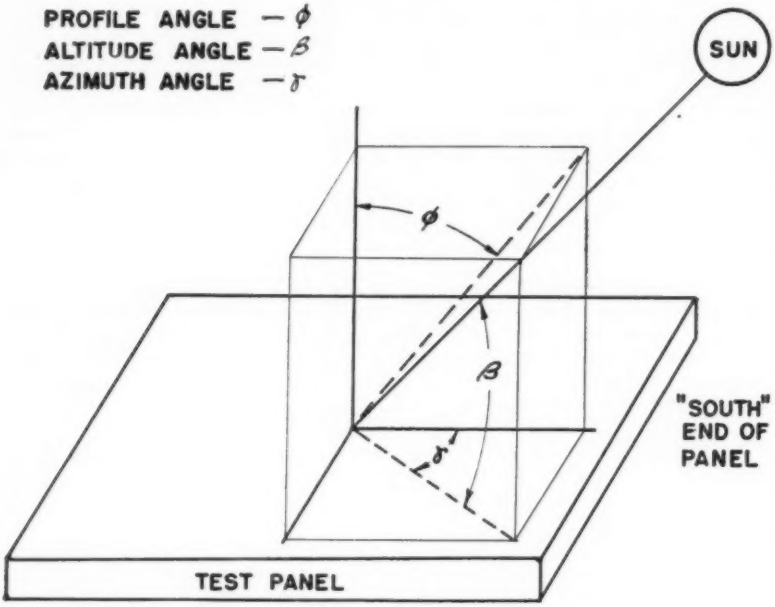


FIG. 3—DEFINITION OF ANGLES

Tests were run by setting the test panel at the desired solar azimuth and altitude with respect to the sun, and rotating and tilting the calorimeter to maintain these fixed angles. In other tests the panel was kept in a horizontal position and the solar altitude was allowed to change while a fixed azimuth was maintained by steadily rotating the apparatus.

The transmitted direct radiation was determined by subtracting the convected and re-radiated gain plus the transmitted diffuse solar energy from the corrected total heat gained by the calorimeter. Percentage of error in these results is of the order of 3 to 5 percent, and is mainly dependent upon the magnitude of the heat leakage corrections.

Results of the tests run on the prism-type glass-block panel at 45 deg increments of solar azimuth are plotted in Fig 4. From the resultant curves and from data obtained from tests run at other azimuth angles the polar coordinate plot of Fig. 5 was made. This provides transmittance values for every sun position.

Similar tests were run on the diffusing-type block panel, but the results, given in Fig. 6, showed no significant dependence on azimuth. Hence, the direct solar transmittance is a function of only the solar altitude.

Most of the tests on the compound louvered device were run at either zero or 180 deg solar azimuth. Previous investigations have shown that the transmittance of direct solar energy by a shading device of this type is dependent only upon the profile angle (defined in Fig. 3) if the long axes of all louvers are parallel and shading by any side panels is small. At zero or 180 deg solar azimuth, the profile angle

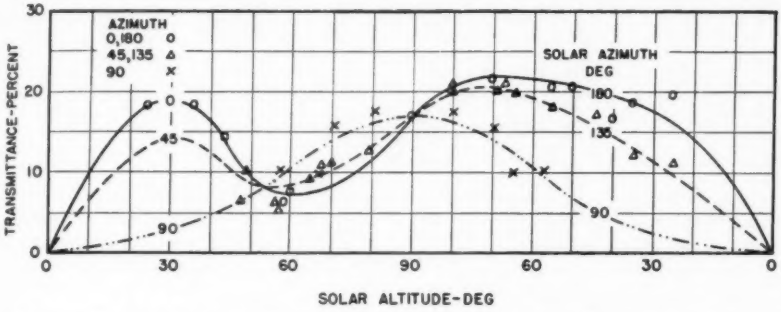


FIG. 4—DIRECT SOLAR TRANSMITTANCE CHARACTERISTICS OF PRISM-TYPE GLASS-BLOCK PANEL

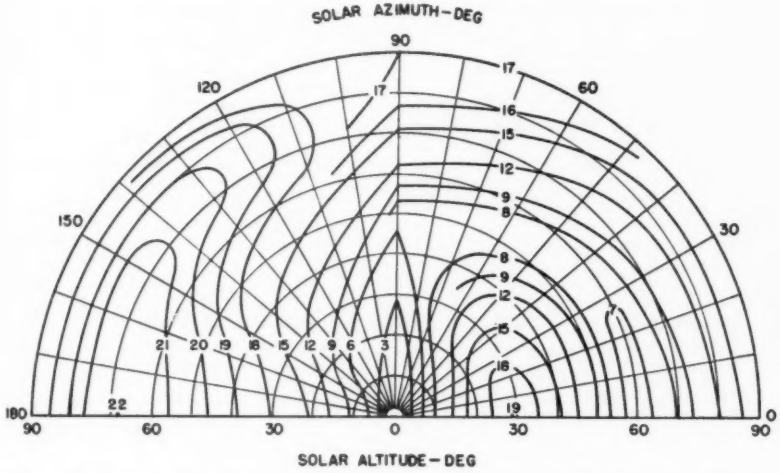


FIG. 5—DIRECT SOLAR ISO-TRANSMITTANCE CURVES FOR PRISM-TYPE GLASS-BLOCK PANEL

becomes simply the complement of the solar altitude. Some tests were run at different solar azimuths to corroborate the absence of an azimuth effect and to study the effect of shading by the side panels. The shading effect was negligible except at a very low solar altitude with the solar azimuth near 90 deg. As incident direct solar radiation is small under these conditions, any corrections can reasonably be neglected.

Fig. 7 shows results for this shading device in conjunction with 2 different types of flat glass. The curve of direct solar transmittances for the louvered device alone was obtained by dividing the direct solar transmittance of each louvered device-glass combination by the diffuse solar transmittance of the glass.

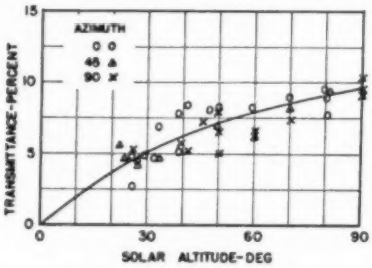


FIG. 6—DIRECT SOLAR TRANSMITTANCE CHARACTERISTICS OF DIFFUSING-TYPE GLASS-BLOCK PANEL

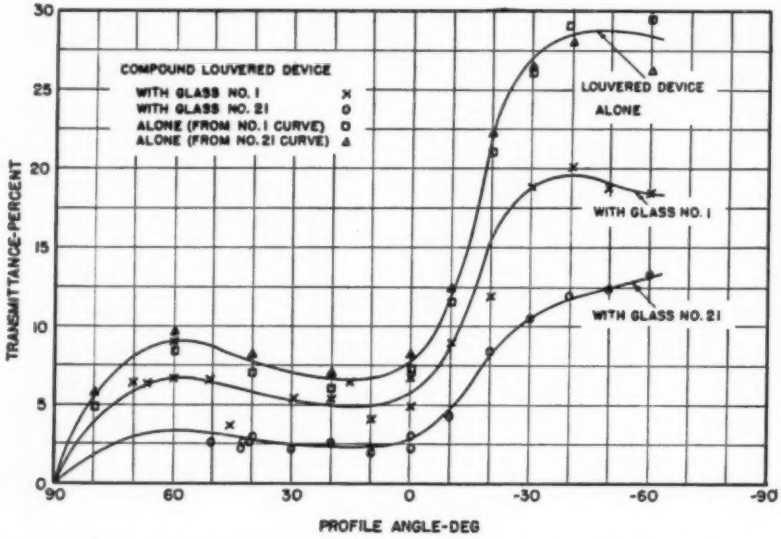


FIG. 7—DIRECT SOLAR TRANSMITTANCE CHARACTERISTICS OF COMPOUND LOUVERED DEVICE

Direct solar transmittance curves for the 3 flat glass fenestrations are shown in Fig. 8 and have been previously reported.[1, 2]

Total Solar Absorptance: Solar absorptances were calculated for the 2 types of glass-block panels from the test data. Because of the approximate nature of the results, no attempt was made to differentiate between diffuse and direct components. The same calculation procedure was used for both panels.

The total absorbed solar radiation was obtained by the following equation:

[Total absorbed solar radiation] = [Rate of heat storage within panel] + [Loss by low temperature radiation exchange at outer surface] + [Convection loss at outer surface] + [Convection and re-radiation loss at inner surface]

This value divided by the total incident radiation gives the solar absorptance.

Heat storage was found by measuring the temperature rise or drop at the 2 surfaces and assuming their average represented the average change within the panel. This value times the heat capacity of the panel divided by the time interval yielded the heat storage rate. The glass-block portion was treated separately from the rest of the panel.

The low-temperature radiation exchange at the outer surface was determined from the incident low-temperature radiation (measured with the radiometer) and the surface temperatures and emissivities of the glass block and concrete.

Convection losses to the outdoors were found by using the following empirical relationship: convection coefficient [Btu per (hour) (sq ft) (Fahrenheit degree)] equals the square root of the wind velocity (miles per hour). Corrections were made

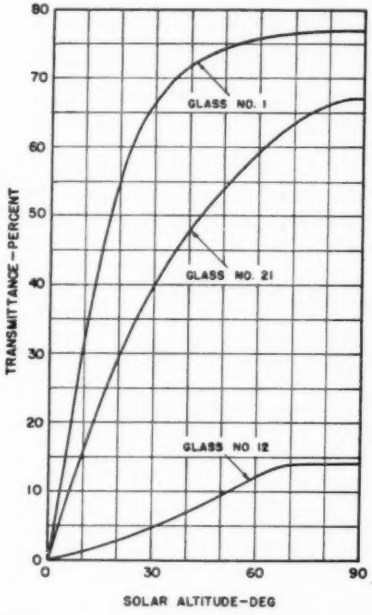


FIG. 8—DIRECT SOLAR TRANSMITTANCE CHARACTERISTICS OF SHEET-GLASS SKYLIGHTS

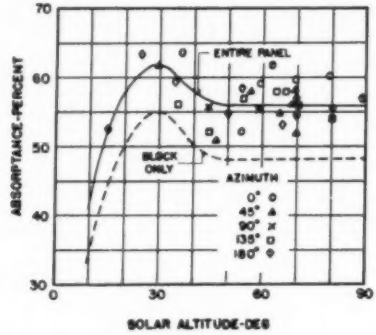


FIG. 9—TOTAL SOLAR ABSORPTANCE CHARACTERISTICS OF PRISM-TYPE GLASS-BLOCK PANEL

for tests when the panel was not horizontal. This formula was determined from data obtained in research work on flat roofs reported recently[4].

Convection and re-radiation exchange between the inner surface and the calorimeter was based on previous calibration work[5] in which an insulated opaque panel was put into the calorimeter opening and the convection gain determined for various temperature differences and orientations. The radiation portion is based simply on the temperatures and emissivities of the surfaces.

The greatest source of error in calculating solar absorptance by the afore-mentioned equation lies mainly in the estimate of the rate of heat storage and to some extent in the estimate of convection loss at the outer surface. It is probable that

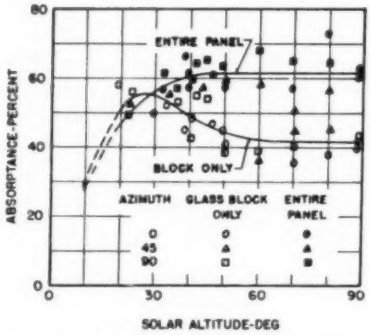


FIG. 10—TOTAL SOLAR ABSORPTANCE CHARACTERISTICS OF DIFFUSING-TYPE GLASS-BLOCK PANEL

the resulting percentage of error is, under some conditions, as great as 10 to 12 percent.

The absorptance curve for the prism-type glass-block panel and glass block alone are presented in Fig. 9. In Fig. 10 are similar curves for the diffusing-type glass-block panel. Results show no apparent solar azimuth effect for any of these curves.

Absorptances for the sheet-glass fenestrations are found in Fig. 11. Values for glass No. 1 are computed values and have been previously reported. Those for figured glasses were determined from experimental work carried on at the Laboratory and are reported here for the first time.

Absorptance values for the louvered device-glass combination were not calculated because of the uncertainty in estimating the radiation and convection losses from the outer surfaces of both the louvered device and the glass.

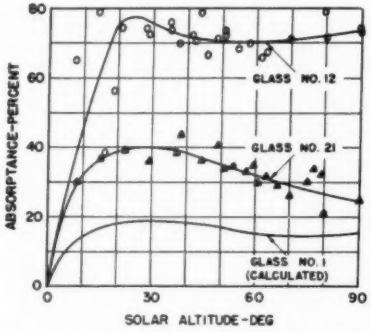


FIG. 11—TOTAL SOLAR ABSORPTANCE CHARACTERISTICS OF SHEET-GLASS SKYLIGHTS

Convection and Re-radiation Gain: Empirical relationships between the total absorbed solar radiation and the convection and re-radiation transfer from the inside of the glass-block and sheet-glass fenestrations are shown in Figs. 12 and 13.

Results for the glass-block panels were obtained from tests during which the panels were held in a fixed horizontal position and subjected to a cyclic input of solar radiation. Because of the heat-storage capacity of the panel, the convection and re-radiation gain lagged behind the absorbed radiation. When these data were compared and the time lag properly taken into account, the straight-line relationships shown in Fig. 12 were obtained. The results for the diffusing-type block are presented for both the entire panel and the glass block only, while for the prism-type panel they are given for the entire panel only. Analysis of the complex heat flow through mortar and reinforcing frame of the latter panel is beyond the scope of this paper and will be treated in a subsequent paper.

The curve (Fig. 13) relating convection and re-radiation gain and absorbed solar radiation for sheet glass was calculated from test data obtained in 1950. Data for a number of different types of glass were reported for various environmental conditions, but in all cases the glass was warmer than the calorimeter surface. Time lag was small and, though it was included, can be neglected in practical load estimat-

ing. This curve applies to any sheet of horizontal glass. A similar curve for vertical glass would show higher convection gains to the calorimeter surface due to higher convection coefficients. Also a vertical panel would have a smaller low-temperature radiation loss to its environment. The curve for sheet glass as well as the curves for the glass-block skylights are corrected to zero air-to-air temperature difference.

It will be noted in Fig. 13 that there is negative convection and re-radiation gain with no absorbed solar radiation and no air-to-air temperature difference. This is due to radiation losses from the glass to the cooler (than ambient air) upper atmosphere.

Test results show a relationship between convection and re-radiation gain and the total solar radiation transmitted by the louvered device and incident upon the glass. This is shown in Fig. 14.

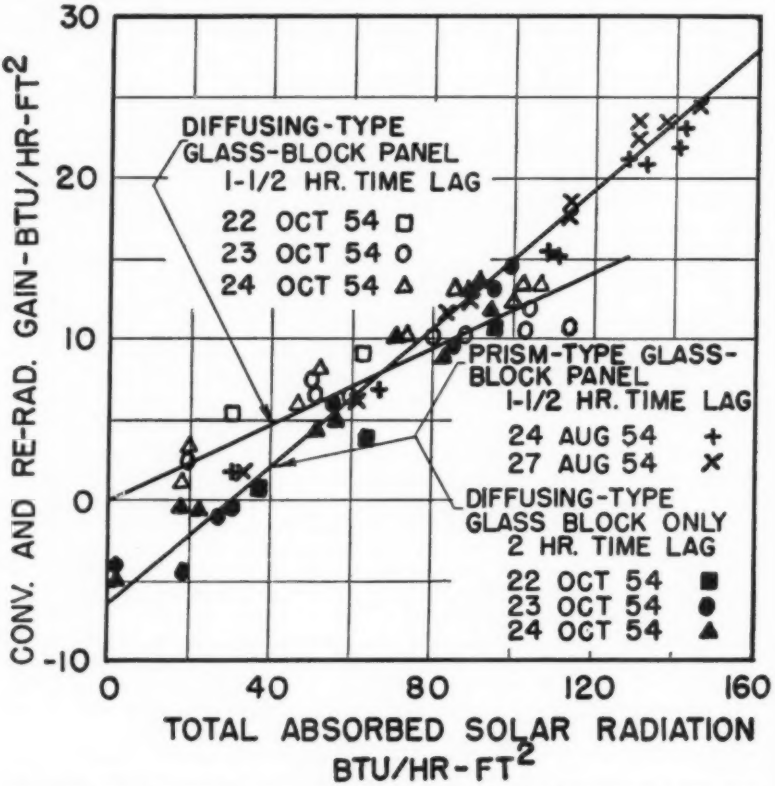


FIG. 12—RELATIONSHIP BETWEEN CONVECTION AND RE-RADIATION GAIN AND ABSORBED SOLAR RADIATION FOR PRISM-TYPE AND DIFFUSING-TYPE GLASS-BLOCK PANELS AND DIFFUSING-TYPE GLASS BLOCK ALONE

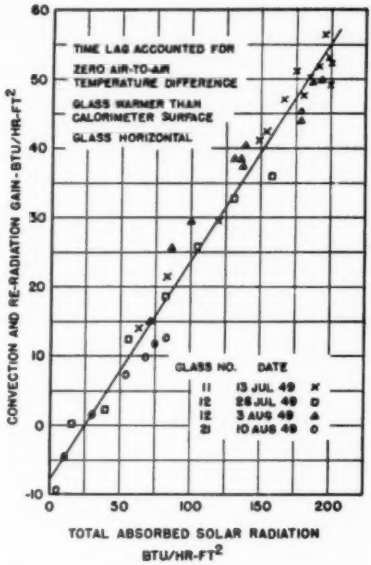


FIG. 13—RELATIONSHIP BETWEEN CONVECTION AND RE-RADIATION GAIN AND ABSORBED SOLAR RADIATION FOR SHEET GLASS

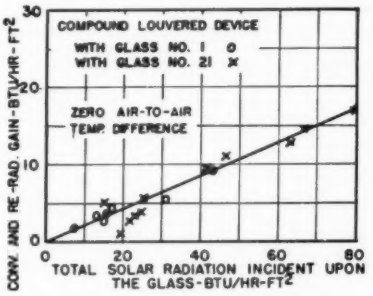


FIG. 14—RELATIONSHIP BETWEEN CONVECTION AND RE-RADIATION GAIN AND TOTAL INCIDENT SOLAR RADIATION UPON THE GLASS FOR COMPOUND LOUVERED DEVICE-GLASS COMBINATIONS

Estimated corrections for each 10 deg air-to-air temperature difference are as follows [for heat flow downward and in Btu per (hour) (square foot) (Fahrenheit degree)]:

A. Prism block panel or diffusing block alone	5
B. Louvered device and glass	8
C. Concrete only	7
D. Sheet glass only	10

These values should not be regarded as a precise means of obtaining heat transmission, as they are intended to give only approximate corrections. The first 2 values were obtained from test data taken during nocturnal periods when the panel and its environment were in near equilibrium. Dividing the corrected heat gain to the calorimeter by air-to-air temperature difference provided the corrections. The values for concrete and sheet glass are calculated.

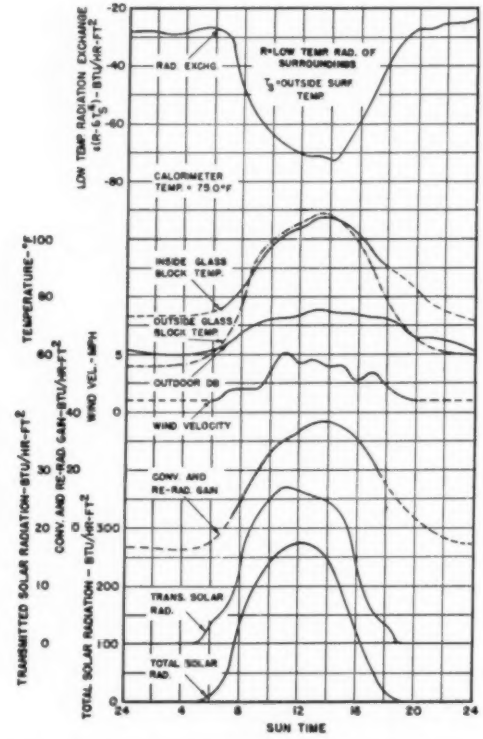


FIG. 15—RESULTS FROM A TYPICAL TEST ON PRISM-TYPE GLASS-BLOCK PANEL—AUGUST 27, 1954

Results from Typical Test Days: Graphs of results for 24-hr fixed-position tests for 3 light-controlling fenestrations are shown in Figs. 15, 16, and 17. Included are the environmental conditions to which the skylight was subjected. A valid comparison between the 3 sets of curves cannot be made due to differences in climatic conditions.

Some idea of the effect of various weather conditions upon the total heat gain can be concluded. From Figs. 15 and 16 it is seen that the convection and re-radiation gain from the glass-block panels is affected largely by the incident solar radiation (actually the absorbed solar radiation). Air-to-air temperature differences appear to have a smaller influence on this portion of the gain, In Fig. 17 it can be seen that the opposite is true for the louvered device and glass fenestration. There appears to be little change in convection and re-radiation gain due directly to incident solar radiation when no direct solar radiation goes directly through the louvers. This portion of the gain, however, is affected largely by the outdoor-air-

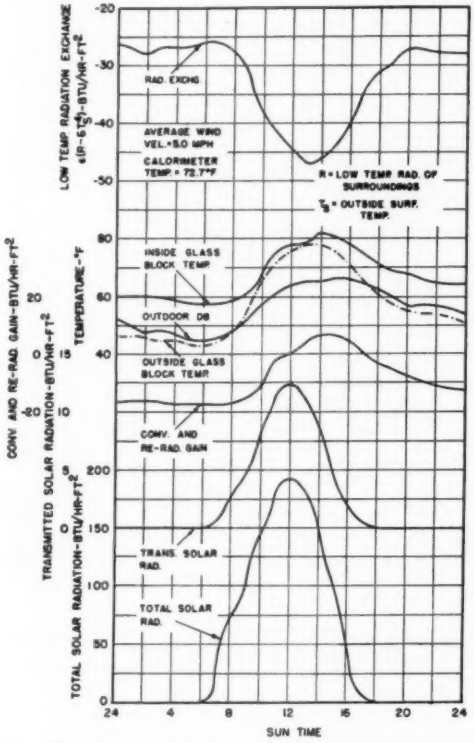


FIG. 16—RESULTS FROM A TYPICAL TEST ON DIFFUSING-TYPE GLASS-BLOCK PANEL—OCTOBER 23, 1954

to-calorimeter-surface temperature difference. These conclusions may be verified by dividing the convection and re-radiation gain by the air-to-air temperature difference at any particular time (remembering for the glass-block panels to take the outdoor air temperature 1½ hr early to account for time lag). During periods

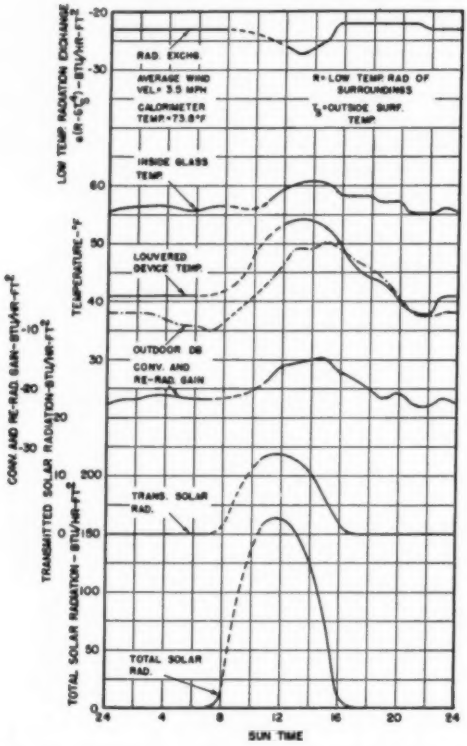


FIG. 17—RESULTS FROM A TYPICAL TEST ON COMPOUND LOUVERED DEVICE WITH REGULAR PLATE GLASS—NOVEMBER 15, 1954

of high solar intensity the convection and re-radiation from the glass-block panels will greatly exceed the amount attributable to conducted heat, while for the louvered device-glass combinations it will exceed it by a smaller amount.

In all 3 graphs, the weather-side surface of the panel becomes cooler than the outdoor air or the calorimeter surface during a portion of the night. This is due to the low-temperature radiation loss to the cool upper atmosphere which frequently becomes greater than the convection gain to the outer surface, especially on clear nights.

Comparison of Total Transmitted Solar Radiation: In Fig. 18 are shown calculated curves of total transmitted solar radiation values for each of the types of skylighting reported. These values are for a design day as specified in the figure. The transmitted radiation characteristics give a good indication of the light-control properties of the various skylights. It is seen that generally the light-control skylight gives a smaller, more uniform transmission of solar energy than the sheet-glass fenestrations. Not apparent from this graph is the *light-diffusing* property of light-controlling skylights.

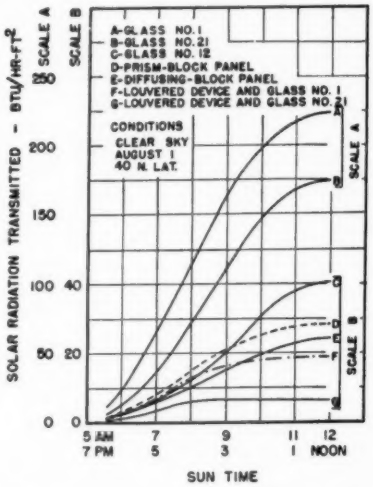


FIG. 18—TOTAL TRANSMITTED SOLAR RADIATION FOR VARIOUS SKYLIGHTING FENESTRATIONS

APPLICATION OF DATA TO PRACTICE

The use of the information in this paper to ascertain the instantaneous cooling load due to various skylighting fenestrations is illustrated by the following 2 typical examples. For simplicity, both examples are for a design day. To determine heat gains for other conditions refer to THE GUIDE, 1955, Chapter 13, for necessary information.

Example 1: *Calculate the instantaneous cooling load due to a* 5 *ft* × 10 *ft*, 18-*block diffusing-type glass-block panel at* 3 *p.m. on August* 1 *at* 40 *deg north latitude; outdoor temperature* = 95 *F, indoor temperature* = 80 *F.*

Solution: *Step* 1. From Table 2, obtain the solar altitude, incident direct and diffuse radiation. These are: solar altitude = 46 deg; direct radiation = 192 Btu per (hr) (sq ft); diffuse radiation = 32 Btu per (hr) (sq ft).

Step 2. Determine portion of panel surface that is glass block. Area of glass blocks equals 18 × 11¾ in. × 11¾ in. × 1/144 (sq ft) or 17.3 sq ft. Total area equals 50

sq ft; therefore, surface is 35 percent glass. The glass block and concrete portions of the panel are treated separately, assuming parallel heat flow.

Step 3. Calculate total transmitted solar radiation through glass block. Diffuse transmittance is 9 percent (Table 1) and direct transmittance 7 percent (Fig. 6) for 46 deg solar altitude and for a panel having 54 percent of the surface area glass block. The total transmitted solar radiation for the glass block alone is then 1/0.54 (0.09 × 32 + 0.07 × 192) or **30.2** Btu per (hr) (sq ft).

Step 4. Find convection and re-radiation from glass block. To take into account the time lag of the block both the absorbed solar radiation and outdoor air temperature

TABLE 2—SUN POSITIONS AND INCIDENT DIRECT AND DIFFUSE SOLAR RADIATION FOR A HORIZONTAL SURFACE

(August 1, clear atmosphere)

LAT.	SUN TIME	SOLAR ALT.	SOLAR AZ.	PROFILE ANGLE	DIRECT RAD.	DIFFUSE RAD.
30 Deg North	6 a.m. 6 p.m.	9.0	106	−60	20	13
	7 5	21.5	99	−22	75	24
	8 4	34.5	92	− 3	138	30
	9 3	47.5	84	6	198	33
	10 2	60.0	73	10	245	34
	11 1	72.0	52	11	272	35
	12 noon	78.0	0	12	284	36
40 Deg North	5 a.m. 7 p.m.	0.5	114	−89	2	1
	6 6	11.5	104	−50	28	16
	7 5	23.0	95	−12	82	25
	8 4	34.5	85	7	138	30
	9 3	45.5	74	15	192	32
	10 2	56.0	59	19	232	34
	11 1	64.5	35	21	257	34
	12 noon	68.0	0	22	265	35
50 Deg North	5 a.m. 7 p.m.	4.5	113	−79	8	6
	6 6	13.5	102	−41	37	18
	7 5	23.5	90	0	85	25
	8 4	33.0	78	18	132	29
	9 3	42.0	65	25	175	32
	10 2	50.0	48	29	208	33
	11 1	56.0	26	31	232	34
	12 noon	58.0	0	32	237	34

must be taken 2 hr previous to 3 p.m. or at 1 p.m. Table 2 gives values of solar altitude = 64.5 deg; incident direct radiation = 257 Btu per (hr) (sq ft); and diffuse radiation = 34 Btu per (sq ft) (hr) at 1 p.m. The absorptance of the glass block from Fig. 10 is 42 percent. This value times the total incident radiation gives a total absorbed solar radiation equal to 122.2 Btu per (hr) (sq ft). From Fig. 12 is then obtained the convection and re-radiation gain due to absorbed solar radiation. This gain is equal to **19.5** Btu per (hr) (sq ft).

The gain due to air-to-air temperature difference must also be calculated. This difference is 15 F (assuming air temperature was 95 F at 1 p.m.), and using the previously

listed correction of 5 Btu per (hr) (sq ft) for each 10 deg difference, a value of **7.5** Btu per (hr) (sq ft) is obtained. The sum of the bold face values yields the gain through the glass-block portion of the panel. This value is 30.2 + 19.5 + 7.5 or 57.2 Btu per (hr) (sq ft).

Step 5. Determine the total gain through the concrete. Refer to Table 3. Listed in this table are approximate values of heat gain for a design day. They are based on equivalent temperature differences obtained from THE GUIDE, 1955, Chapter 13, Table 9, and a U-value of 0.7 Btu per (hr) (sq ft) (F deg) for the concrete. The gain for 3 p.m. is found to be 36 Btu per (hr) (sq ft).

TABLE 3—HEAT GAIN THROUGH CONCRETE PORTION OF DIFFUSING-TYPE GLASS-BLOCK PANEL FOR DESIGN DAY[a]

[*Btu per (hour) (square foot)*]

TIME	HEAT GAIN	TIME	HEAT GAIN	TIME	HEAT GAIN
8 a.m.	0	1 p.m.	31	7	22
9	2	7	35	8	15
10	14	3	36	9	12
11	21	4	36	10	8
12 noon	27	5	32	11	6
		6	28	12 mid.	4

[a] Values obtained using THE GUIDE, 1955, Chapter 13, Table 9, and U value equal to 0.7 Btu (hr) (sq ft) (F deg).

Step 6. Combine the gain through the glass block and concrete in proper proportion. 0.35 × 57.2 + 0.65 × 36 = 43.4 Btu per (hr) (sq ft) of panel, or 2170 Btu per hr for the entire panel.

Example 2. *Determine the instantaneous heat gain per square foot on August* 1, 50 *deg north latitude at* 10 *a.m. for the compound louvered device-glass No.* 1 *combination. Assume no air-to-air temperature difference.*

Solution: *Step* 1. Determine the profile angle, incident direct solar radiation and diffuse radiation from Table 2. These are: profile angle = 29 deg; direct radiation = 208 Btu per (hr) (sq ft); diffuse radiation = 33 Btu per (hr) (sq ft).

Step 2. Find the total transmitted solar radiation. The direct transmittance is 5 percent (Fig. 7) and the diffuse transmittance, 11 percent (Table 1). Multiplying the values of incident radiation by the proper transmittances, 0.05 × 208 + 0.11 × 33, yields the total transmitted solar radiation. This value is **14.0** Btu per (hr) (sq ft) of panel.

Step 3. Determine the convection and re-radiation gain from inside of glass. First the incident solar radiation upon the *glass* must be found. This can be done by multiplying the two components of solar radiation by the appropriate transmittances of the louvered device *alone*. For a profile angle of 29 deg, the direct transmittance is 0.07 and the diffuse transmittance 0.16 (Table 1). The solar radiation incident upon the glass becomes 0.07 × 208 + 0.16 × 33 or 19.8 Btu per (hr) (sq ft). Using Fig. 14 it is seen that the gain is then **4.1** Btu per (hr) (sq ft).

Step 4. The total gain per square foot is equal to the sum of the bold face values: 14.0 + 4.1 = 18.1 Btu per (hr) (sq ft).

CONCLUSIONS

Accurate means of determining the total transmitted solar energy have been given, together with an approximate method for the convection and re-radiation

gain. The values obtained represent the instantaneous cooling load. Development of a thermal-circuit representation of glass-block panels is in the process and will bring forth more accurate means of computing the convection and re-radiation gain.

ACKNOWLEDGEMENT

The authors wish to thank the members of the TAC on Heat Flow through Glass for their assistance in conducting this program, with special thanks to W. B. Ewing, Ewing Development Co.; D. D'Eustachio, Pittsburgh Corning Corp.; and H. B. Vincent, Owens-Illinois Glass Co. Credit for much of the test work and calculations goes to the college students who assisted with this project; J. H. Malone, L. Holstein and T. J. Vild.

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2. ASHVE RESEARCH REPORT No. 1417—Solar Energy Transmittance of Figured Rolled Glass, by G. V. Parmelee and W. W. Aubele (ASHVE TRANSACTIONS, Vol. 57, 1951, p. 209).

3. ASHVE RESEARCH REPORT No. 1485—Design Data for Slat-Type Sun Shades for Use in Load Estimating, by G. V. Parmelee and D. J. Vild (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 403).

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DISCUSSION

R. W. MCKINLEY, Pittsburgh, Penna.: As always, I believe that the researcher will serve his audience best if he presents his conclusions and their applications and limitations first and follows these by a description of methods and a discussion of results.

We find in this field that too often the people who perhaps *should* dig deeply find neither time nor incentive; though, I believe, incentive is building up.

Similarly, I would suggest that Figs. 2, 3, and 12 in the final printing be enlarged and then relettered to make them more readily legible. In their present form, I am afraid that they are not only difficult to read, but in one case at least, hard-to-understand unless you are quite familiar with the basic material.

Commenting on Fig. 18, which really is a comparison of the different types of skylights, I would say that it is very useful and most important from an application point of view. I would like to ask the authors if it would not be a better paper if in addition to Fig. 18, or perhaps in place of Fig. 18, a somewhat similar plot were made of *total instantaneous heat gain* for a similar design day. Because this plot is at the tail end of the paper, some people may overlook its importance.

I don't know how much control they have over the space available, but if space can be afforded I would like to suggest that the authors provide some summary indication of the surface temperature record. Some surface temperatures were indicated for each of the panels, for a single day, but not on a directly comparable basis. Comparable surface temperatures would be of interest to people who may wish to study the mean radiant temperature relative to comfort.

Although I agree with the authors that perhaps the greater importance and certainly the most popular interest is in the cooling season aspects of skylight performance, certainly there is importance related to their performance during the heating season. I think that some mention should be made and some data provided so that heating season performance can be evaluated.

The authors refer to the relationship between the transmittance data and the illumination performance of the panel. I think this is a worthy contribution to both the illumination and heat flow fields. Someone ought to take up the point and develop it further, at least to the extent of suggesting a rough correlation or conversion factor between illumination and heat flow.

E. W. CONOVER, Detroit, Mich.: I have but one comment to make on this paper and not as a member of the Committee, but from my personal experience. I believe that the results are all based on what we might term to be ideal conditions. By that I mean that the glass surfaces are clean.

Early in the research work of my company we found that dirt collection both inside and out, particularly on horizontal surfaces, had a tremendous effect on the light transmission in urban and in industrial areas. I am wondering if it would be possible to find out what effect dirt collection would have on heat gain through these skylight panels.

R. A. MILLER, Pittsburgh, Penna.: This is certainly unexpected. I had no idea or intention of coming up and saying anything. The work that has been done at the Laboratory on skylight fenestration has been pursued actively and carefully with the results as shown here. I feel that the value of the work still has to be completed by getting together data that can be translated into total cooling load and total heating load; and there is a great deal of available information which can be so tied in ultimately when the opportunity can be taken.

With respect to the question of the effect of dirt on the transmittance, that is beyond any dispute. To determine what the dirt covers and how you are then going to evaluate it, is something else again. If one takes a value for transmittance and then by varying stages of cleaning the sample, determines further transmittances under dirt conditions, you can certainly get values.

To put the glass out in the atmosphere and let it gather dirt for a certain length of time and then evaluate it gives you that particular value, but it is not applicable to any other piece of glass exposed for any other period of time or the same period of time at some different place.

The only way to arrive at values is to use the cleanest glass you can get and then if you wish to modify, substitute other values for the transmittances in the tabulation which you have arrived at. I believe that covers Mr. Conover's suggestion.

AUTHORS' CLOSURE, (Mr. Vild): My co-author and I would like to thank Mr. McKinley for his suggestions concerning the presentation and writing of technical papers. There are many members who concur with his ideas and, should they represent the popular demand, I am sure changes will be made. The size of figures and tables included in the paper is out of the hands of the authors so we can make no comment on that subject.

Fig. 18 was meant to show the relationship of light transmittances as well as solar transmittances. For this reason the convected and re-radiated gain was excluded. It is agreed that a plot of total instantaneous heat gain would be helpful. However, space limitations make it difficult to include additional material of limited interest. We have included all design data required to obtain instantaneous cooling loads for a design data. Information pertinent to comfort determination was deemed outside the scope of this paper.

The instantaneous heating load due to each type of fenestration can be found using information in Table 20, Chapter 9, 1956 ASHAE GUIDE.

Mr. Conover has posed a problem that is a difficult one to correlate. In determining the effect of dirt upon the performance of any fenestration, the big problem is in defining the quantity of dirt. We have run a few tests on glass samples exposed to the weather. The results mean little since they would not be applicable to any other sample. Dirt would decrease the total instantaneous cooling load due to a particular fenestration. The transmitted solar energy would decrease more than the convected-and-reradiated gain would increase due to the increased absorptance of the surface.

This test program was another step in the extended ASHAE program to provide its members with reliable design data for all types of daylighting fenestrations. Many types, however, still require investigation. It is hoped that this much-needed information will soon be obtained through ASHAE research.

1561

No. 1561

MOISTURE IN TRANSIENT HEAT FLOW

By K. R. SOLVASON*, OTTAWA, CANADA

CALCULATION of building heat losses and gains is normally based upon coefficients obtained for dry materials. The presence of moisture, when taken into account, is usually considered to influence the heat transmission by increasing the thermal conductivity. Moisture, according to this simple concept, introduces little additional complication into the calculation of heat flow if it remained stationary within the materials involved. Hutcheon and Paxton[1] have shown that in a hygroscopic material such as wood sawdust the moisture migrates under changing temperature conditions, thus providing a substantial additional mechanism of heat transmission.

Hutcheon and Paxton[1] demonstrated that for wood sawdust, the final redistribution of moisture following imposition of a fixed temperature gradient on the moist sample, could be predicted from the known hygroscopic properties of the sawdust. Following their method the moisture redistribution throughout a sealed sample of sawdust containing 8 percent moisture by weight, has been calculated, and is shown in Fig. 1, for a change from a uniform temperature of 60 F throughout, to 100 F one side and 60 F on the other.

If the sample thickness is 4 in. and the density of dry material is 6 lb per cu ft, approximately 0.022 lb of water per sq ft of sample area must be evaporated in the region from $x/L = 0$ to $x/L = 0.6$, migrate as vapor, and be condensed in the region from $x/L = 0.6$ to $x/L = 1$.

Assuming the latent heat of evaporation to be 1000 Btu per lb, a heat flow of 22 Btu per sq ft across the plane $x/L = 0.6$ results because of the vapor flow.

Once redistributed, the moisture, at least in sawdust, adds little to the heat transfer rate for the dry material During the redistribution, however, the heat transmission is increased because of the vapor transfer and there is a marked deviation from the conditions for the corresponding case of dry materials.

This mechanism and its effects have now been further examined for several combinations of materials and temperatures.

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* Assistant Research Officer, Division of Building Research, National Research Council.

[1] Moisture Migration in a Closed Guarded Hot Plate, by N. B. Hutcheon and J. A. Paxton (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 301).

Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

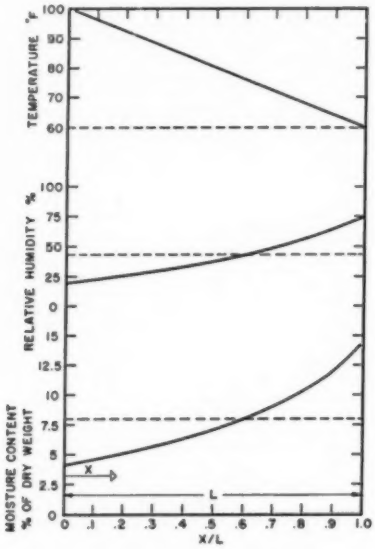


FIG. 1—TEMPERATURE, RELATIVE HUMIDITY, AND MOISTURE DISTRIBUTION IN WOOD SAWDUST WITH AVERAGE MOISTURE CONTENT OF 8 PERCENT FOR $T_1 = T_2 = 60$ F (BROKEN LINE) AND $T_1 = 100$ F, $T_2 = 60$ F (FULL LINE)

TESTS

Samples of a number of materials, singly and in combination, 7 in. square and of thicknesses up to 2 in. were inserted between parallel heated and cooled plates. Tests were carried out with the materials dry and then with moisture added. Transient conditions were established by suddenly changing the temperature of one plate, and heat flow rates and temperatures were recorded throughout the test period until conditions had substantially reached steady state.

The materials tested included the following composite and homogeneous samples:

Test No. 1: In this test preformed glass wool was placed between 2 pieces of fiberboard ½ in. thick and with a density of 15.25 lb per cu ft, or a total weight 1.27 lb per sq ft of area. The glass wool was ⅞ in. thick and had a density of 11.1 lb per cu ft, or a total weight 0.81 lb per sq ft. The moisture contents tested were zero and approximately 5.1 percent of the dry weight of fiberboard or a total of 0.065 lb per sq ft.

Test No. 2: In this test preformed glass wool was placed between 2 sheets of blotting paper 0.023 in. thick and with density 33.1 lb per cu ft. The total weight of paper was 0.127 lb per sq ft. The glass wool had a thickness of ⅞ in. and a density of 12.4 lb per cu ft. The moisture contents were zero, 0.0325 and 0.065 lb per sq ft.

Test No. 3: In this test the material used was expanded vermiculite, screened through a 13-mesh screen, having a thickness of 1¼ in. and a density of 16.3 lb per cu ft, or a

weight of 1.70 lb per sq ft. Moisture contents of zero and 0.065 lb per sq ft of area or 3.8 percent of the dry weight of material were tested.

Test No. 4: The material used was spruce sawdust, 2.156 in. thick having a density of 12.0 lb per cu ft, or a weight of 2.656 lb per sq ft. The moisture contents were zero and 9 percent of the dry weight or 0.194 lb per sq ft of area.

APPARATUS AND TEST PROCEDURE

The testing apparatus developed by Hutcheon and Paxton and modified slightly for these tests, is shown in Fig. 2. It consisted of two 7-in. square hollow aluminum plates between which the test samples were placed. Water at a controlled temperature was circulated through a channel in the plates to produce the desired plate temperatures. Multiple thermocouple bakelite heat meters 4½ in. square by 3/64 in. thick were cemented to the central portions of the plates to measure the heat flows at the boundaries. Strips of bakelite 3/64 in. thick were cemented to the remaining area of the plates surrounding the meters to provide a guard area around the meters. Thermocouples embedded in the plate surface were used to measure plate temperatures.

The hot plate temperature was maintained by circulating water from an electrically heated 5-gal reservoir. The cold plate was supplied with water from a refrigerated 60-gal reservoir. A valve arrangement permitted either hot or cold water to be circulated through both plates simultaneously or hot water through one and cold water through the other. Water temperatures had to be accurately controlled since a rapid change in water and plate temperature even though of very small amplitude would cause a large change in the heat meter voltage. A commercial electronic temperature controller was adapted to provide a modulating form of control for both reservoirs.

In tests Nos. 1, 2 and 3, samples with thermocouples spaced at intervals as shown in Fig. 2 were sealed in polyethylene bags and placed between the plates. Cold water (temperature T_2 = 60 F approximately) was circulated through both plates until the temperature at all planes reached plate temperature. Hot water

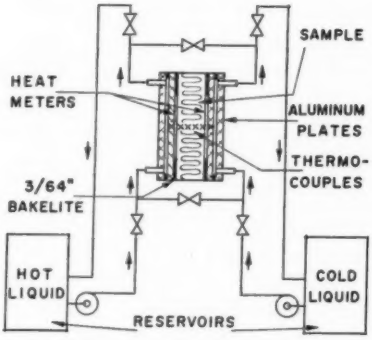


FIG. 2—TESTING APPARATUS — BOTH LIQUID RESERVOIRS ARE PROVIDED WITH ELECTRONIC TEMPERATURE CONTROL

(temperature T_1 = 100 F approximately) was then substituted for the cold water in the plate at $x = 0$. Starting at the time of the temperature change ($\theta = 0$) readings of the heat meter voltages and of the temperatures were taken at definite time intervals until very nearly steady state conditions prevailed. The electronic temperature indicator used was suitably modified to give heat meter voltages as well as temperatures.

The temperatures T_1 and T_2 were maintained for sufficient time to ensure steady state conditions and the conductance for the dry sample was obtained. The temperature at the plate $x = 0$ was then suddenly reduced to T_2 and heat flows and temperatures again recorded.

With the sample at temperature T_2 throughout, a measured amount of water was introduced near the plate $x = 0$, without disturbing the plate spacing or the sample in any way, and sufficient time was allowed to permit the water to become distributed. The transient heat flow and temperature relationships for the sample were again recorded.

Test No. 4 was performed using an apparatus similar to that used in tests 1, 2 and 3 but with provision to control the plate temperatures from 10 F to 80 F. Heat flow was measured at the $x = 0$ boundary only. The plate temperatures for this test were T_2 = 41 F, T_1 increased from 41 F to 79 F; T_1 = 79 F, T_2 increased from 41 F to 79 F; and T_2 = 12 F, T_1 increased from 12 F to 47 F.

THEORETICAL HEAT FLOW

The temperature and heat flow relationships following such sudden boundary temperature changes can be expressed by solutions of the Fourier equation:

$$\delta T/\delta\theta = \alpha\,(\delta^2 T/\delta x^2),$$

where

T = temperature (F).
θ = time (hours).
x = distance from the reference boundary (feet).
$\alpha = k/\rho c$, the thermal diffusivity, square feet per hour.
k = thermal conductivity, Btu per (hour) (square foot) (Fahrenheit degrees per foot).
ρ = density, pounds per cubic foot.
c = specific heat, Btu per (pound) (Fahrenheit degree).

If the temperature is initially T_2 at all planes and at time $\theta = 0$ the temperature at $x = 0$ is suddenly increased to T_1, while at $x = L$ the temperature remains at T_2, the temperature is given by:

$$T = T_2 + (T_1 - T_2)\left[1 - x/L - 2\sum_{n=1}^{\infty}\frac{1}{n\pi}\,e^{-n^2 z\theta}\,\sin\frac{n\pi x}{L}\right]. \quad . \quad . \quad . \quad (1)$$

where

$$z = \pi^2\alpha/L^2.$$

The heat flow at $x = 0$ will be given by

$$Q_0 = \frac{K}{L}(T_1 - T_2)\left[1 + 2\sum_{n=1}^{\infty} e^{-n^2 z\theta}\right],$$

which can be written

$$Q_0 = Q_S[1 + S(z\theta)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

Q_S = the final steady state heat flow which is seen to equal $K/L(T_1 - T_2)$.
$S(z\theta)$ = the series,

$$2\sum_{n=1}^{\infty} e^{-n^2 z\theta}$$

Similarly at $x = L$ the heat flow will be

$$Q_L = \frac{K}{L}(T_1 - T_2)\left[1 - 2\sum_{n=1}^{\infty}(-1)^{n+1} e^{-n^2 z\theta}\right]$$

or

$$Q_L = Q_S[1 - B(z\theta)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If after steady state conditions have been attained with temperatures T_1 at $x = 0$ and T_2 at $x = L$, *i.e.* $T = T_2 + (T_1 - T_2)\ (1 - x/L)$, the temperature at $x = 0$ is suddenly reduced to T_2, the temperature and heat flow relations are given by:

$$T = T_2 + (T_1 - T_2)\left[\sum_{n=1}^{\infty} \frac{2}{n\pi} e^{-n^2 z\theta} \sin \frac{n\pi x}{L}\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$Q_0 = -Q_S[S(z\theta)]. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$Q_L = Q_S[B(z\theta)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

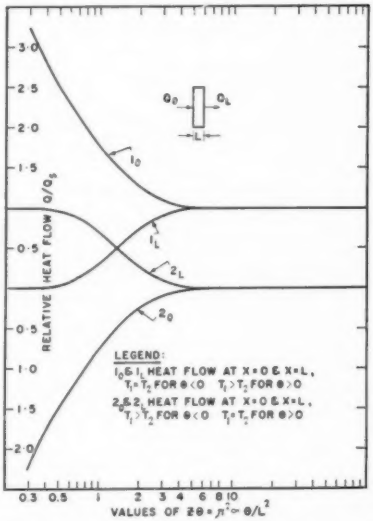


FIG. 3—THEORETICAL HEAT FLOW *vs.* $\pi^2\alpha\theta/L^2$ FOLLOWING SUDDEN TEMPERATURE CHANGES AT $x = 0$

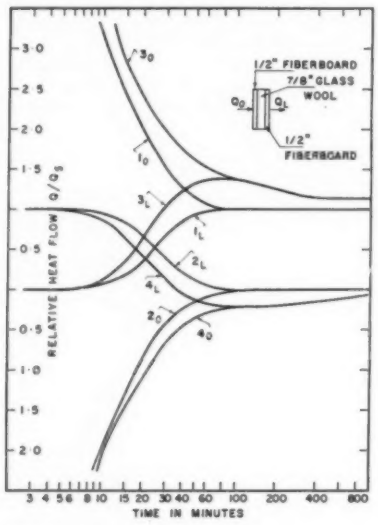
where

Q_S = the steady state heat flow before time $\theta = 0$.
$S(z\theta)$ and $B(z\theta)$ = the series as in Equations 2 and 3.

Typical values of Q_0/Q_S and Q_L/Q_S, which will hereinafter be referred to as the relative heat flows, are plotted against ($z\theta = \pi^2 \alpha\theta/L^2$ in Fig. 3, using calculated values of S ($z\theta$) and tables of B ($z\theta$).

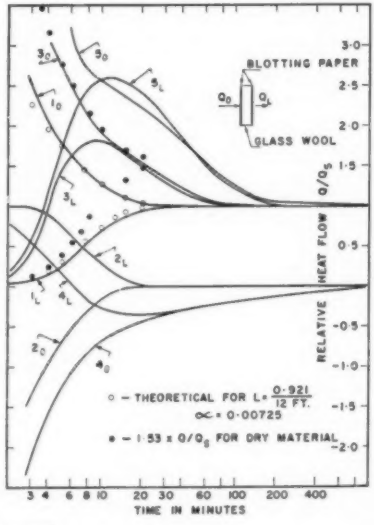
Experimental heat flow values may deviate from the theoretical values due to departure from unidirectional heat flow and to the finite time required to produce a boundary temperature change.

The equations may, however, be used as a check on experimental values for dry materials and as a basis of comparison for the corresponding observations with moist materials. They may also be used to examine the extent to which the case with moisture present may be represented by the case for the dry material with adjustments to the coefficients.



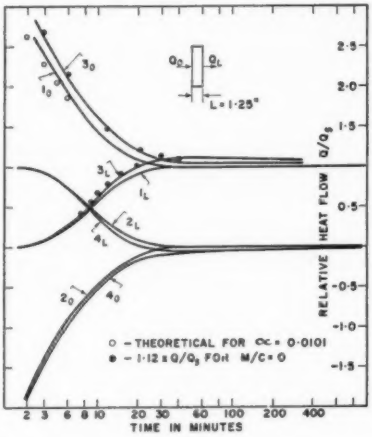
LEGEND
1_0 & 1_L: M/C = 0
3_0 & 3_L: M/C = 0.065 lb/ft³
T_2 = 60F T_1 increased 60F to 100F at $\theta = 0$
2_0 & 2_L: M/C = 0
4_0 & 4_L: M/C = 0.065 lb/ft³
T_2 = 60F T_1 decreased 100F to 60F at $\theta = 0$
Final Q/Q_S for 3_0 & 3_L = 1.05
Q_S = 6.72 Btu/hr for ΔT = 40F

FIG. 4—HEAT FLOW *vs.* TIME FOR GLASS WOOL AND FIBERBOARD COMBINATION FOLLOWING SUDDEN BOUNDARY TEMPERATURE CHANGES (TEST NO. 1)


BLOTTING PAPER
GLASS WOOL
RELATIVE HEAT FLOW Q/Q_S
○ — THEORETICAL FOR L = 0.921/12 FT.
∝ = 0.00725
• — 1.53 x Q/Q_S FOR DRY MATERIAL
TIME IN MINUTES

LEGEND
1_0 & 1_L: M/C = 0
3_0 & 3_L: M/C = 0.0325 lb/ft³
5_0 & 5_L: M/C = 0.065 lb/ft²
T_2 = 60F T_1 increased 60F to 100F at $\theta = 0$
2_0 & 2_L: M/C = 0
4_0 & 4_L: M/C = 0.065 lb/ft²
T_2 = 60F T_1 decreased 100F to 60F at $\theta = 0$
Final Q/Q_S for 5_0 & 5_L = 1.02
Q = 10.8 Btu/hr for ΔT = 40F

FIG. 5—HEAT FLOW *vs.* TIME FOR GLASS WOOL AND BLOTTING PAPER COMBINATION FOLLOWING SUDDEN BOUNDARY TEMPERATURE CHANGES (TEST NO. 2).



LEGEND
1_0 & 1_L: M/C = 0
3_0 & 3_L: M/C = 0.065 lb/ft³
T_2 = 60F T_1 increased 60F to 100F at θ = 0
2_0 & 2_L: M/C = 0
4_0 & 4_L: M/C = 0.065 lb/ft³
T_2 = 60F T_1 decreased 100F to 60F at θ = 0
Final Q/Q_S for 3_0 & 3_L = 1.025
Q = 15.2 Btu/hr for ΔT = 40F

FIG. 6—HEAT FLOW *vs.* TIME FOR EXPANDED VERMICULITE FOLLOWING SUDDEN BOUNDARY TEMPERATURE CHANGES (TEST NO. 3)

DISCUSSION OF RESULTS

The relative heat flows Q_0/Q_S and Q_L/Q_S (Q_S = the steady state heat flow for dry material) are presented graphically in Figs. 4, 5, 6 and 7, for tests Nos. 1, 2, 3 and 4 respectively. The temperature rise with time at various planes in test No. 2 is shown in Fig. 8. A logarithmic time scale is used, in all cases, as a convenient base for plotting.

The presence of moisture, as was expected, very markedly increased the transient heat flows occurring when T_1 at $x = 0$ was increased. An increase in heat flow occurred not only at the $x = 0$ boundary but also at the $x = L$ boundary. Maximum relative heat flows Q_L/Q_S of 1.4 (Fig. 4), 2.6 (Fig. 5) and 1.3 (Fig. 6) were observed in tests 1, 2 and 3 respectively. These values, according to the theory for dry materials, (Equation 3) should have approached but not exceeded the final steady state values of 1.05, 1.02 and 1.025 respectively which were found with moisture present.

The time required to reach a given value of Q_L/Q_S was considerably reduced with moisture present. Comparative times, for example, to reach $Q_L/Q_S = 0.9$ were 27/44, 3.5/16 and 15/19 for tests Nos. 1, 2 and 3 respectively.

It may be noted that the results just presented for the case of an increase in

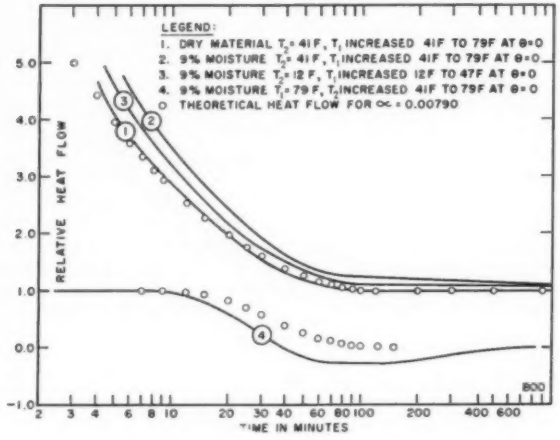

LEGEND:
1. DRY MATERIAL T_2 = 41 F, T_1 INCREASED 41F TO 79F AT θ=0
2. 9% MOISTURE T_2 = 41 F, T_1 INCREASED 41F TO 79F AT θ=0
3. 9% MOISTURE T_2 = 12 F, T_1 INCREASED 12F TO 47F AT θ=0
4. 9% MOISTURE T_2 = 79 F, T_1 INCREASED 41F TO 79F AT θ=0
THEORETICAL HEAT FLOW FOR ∝ = 0.00790
RELATIVE HEAT FLOW
TIME IN MINUTES
800

FIG. 7—HEAT FLOW *vs.* TIME FOR SPRUCE SAWDUST FOLLOWING SUDDEN BOUNDARY TEMPERATURE CHANGES (TEST NO. 4)

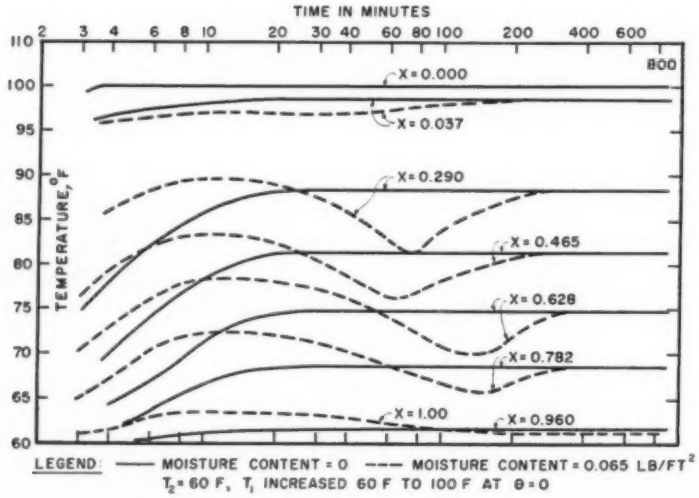

TIME IN MINUTES
800
X = 0.000
X = 0.037
X = 0.290
X = 0.465
X = 0.628
X = 0.782
X = 1.00
X = 0.960
TEMPERATURE, °F
LEGEND: —— MOISTURE CONTENT = 0 ---- MOISTURE CONTENT = 0.065 LB/FT²
T_2 = 60 F, T_1 INCREASED 60 F TO 100 F AT θ = 0

FIG. 8—TEMPERATURE RISE *vs.* TIME AT VARIOUS PLANES FOR GLASS WOOL BLOTTING PAPER COMBINATION (TEST NO. 2)

T_1 from 60 F to 100 F, T_2 remaining at 60 F, are for the first such operation, following introduction of the water near the $x = 0$ plate. The corresponding curves obtained for subsequent cycles produced by reducing T_1 to 60 F and then increasing to 100 F a second time did not show exactly the same moisture effects as the initial curves. The influence of moisture was generally less. The moisture, introduced near $x = 0$, although given some considerable time to seek a uniform distribution was prevented from reaching this condition because of the hysteresis effect in adsorption and desorption, so that the first operation following addition of the water always began with a higher moisture content near the $x = 0$ side.

The curves presented show the instantaneous rates of heat flow. It is of interest also to consider the effect of moisture in terms of the total additional heat transmitted in a given period. This can be found from the area between the curves for the wet condition and that for the dry test. The cumulative increase in heat flows at the $x = L$ plate, for various times after increase of T_1, are given in Table 1 for the case of T_1 increased from 60 F to 100 F. The results for test No. 4 with sawdust are not included since heat flow was metered only at $x = 0$.

It may be noted that the corresponding latent heat transmission, if all the water were evaporated at $x = 0$, moved as vapor and condensed at $x = L$, would be 32 and 65 Btu for the 2 moisture contents. The increased heat flow at the end of 6 hr varied from 16 percent of this latent heat value for test No. 1 with the fiberboard and glass wool sample to 47 percent in test No. 2 for the blotting paper and glass wool sample. However, in test No. 1 the cumulative increase in heat flow was still growing and at the end of 20 hr had amounted to 36 percent of the latent heat value.

The effects of moisture on the heat flows in the cases in which the temperature T_1 at the $x = 0$ boundary was reduced from 100 F to 60 F, T_2 at $x = L$ remaining constant at 60 F, were of the same order as those occurring upon an increase in T_1 from 60 F to 100 F as just discussed. The heat flow *out* at $x = 0$ following a decrease of T_1 was increased by the presence of moisture. The heat flow *out* at $x = L$, which in the dry case decreased gradually from the steady state value to zero, was caused by the presence of moisture to decrease much more rapidly, then to become zero and to be followed by a heat flow *in* which reached a maximum value and then reduced to zero. The heat flow *in* reached values of Q_L/Q_S of 0.21, 0.34 and 0.05 for tests Nos. 1, 2, and 3 respectively.

During the transient period, the net heat flow *out* at both sides must always equal the difference in heat stored in the material between initial and final conditions. Without moisture present, heat flowed out at both sides, gradually re-

TABLE 1—INCREASE IN HEAT FLOW IN BTU AT THE $x = 0$ PLATE AT VARIOUS TIMES FOLLOWING AN INCREASE IN T_1 FROM 60 TO 100 F

TEST NO.	MOISTURE CONTENT LB/SQ FT	TIME FROM INCREASE OF T_1 (HR)					INCREASE AT 6 HR IN % OF LATENT HEAT VALUE
		0.5	1	3	6	20	
1. Fiberboard and glass wool	0.065	0.5	1.7	6.2	10.6	23.7	16
	0.032	6.9	9.6	12.0	13.0	13.9	43
2. Blotting paper and glass wool	0.065	9.2	15.6	24.7	27.9	30.4	47
3. Expanded vermiculite	0.065	0.64	5.8	9.1	12.7	24.6	38

ducing to zero as temperatures equalized. However, with moisture present, which was redistributed by a process of evaporation, vapor flow and condensation as the temperatures equalized, heat had to be provided for the evaporation and absorbed upon condensation. The heat of evaporation was supplied partly from the plate at $x = L$ and partly by conduction from the condensing region. The balance of the heat liberated upon condensation moved by conduction to the plate at $x = 0$. The heat required for evaporation was first drawn from the sensible heat in the material, but as the evaporation continued, the temperature was reduced below 60 F, resulting in a heat flow *in* at $x = L$.

During the redistribution, moisture evaporated at one plane need not necessarily migrate to another plane and remain there, but may successively be condensed and re-evaporated at several intermediate planes. This phenomenon is illustrated by the temperature curves for test No. 2 (Fig. 8). The rapid temperature rise to the first maximum indicates condensation which is followed by a temperature drop indicating re-evaporation and finally a slow rise approaching the value for the dry material as the evaporation is completed. The amount of moisture present in this test was more than could be absorbed hygroscopically, leading to rather high humidities, so that this effect was more pronounced than would normally be expected.

A mathematical treatment of the case with moisture present would be complicated since the conductive heat flow and the latent heat flow are so inter-related. The latent heat flow must operate both in series and in parallel with the conductive heat flow, since heat must flow by conduction to the plane at which water is being evaporated, be transported in the vapor to the plane at which water is being condensed and again flow by conduction to the cold boundary. Thus, in effect, the heat flow can bypass the thermal resistance between the evaporating and condensing regions.

During part of the transient with moisture present the heat flow at $x = 0$, appears, in some of the tests, to be a constant multiple of the heat flow for the dry material. Points have been plotted in Figs. 5 and 6 showing 1.53 and 1.12 times the heat flow for the dry material. These agree fairly closely with heat flow for the moist material for the time from $\theta = 0$ until steady state conditions were reached with the dry material. The $x = 0$ heat flow with moisture can be represented approximately by equation 2 using a lower value for α. This device does not, however, usefully describe the heat flow at $x = L$ since Q_L/Q_S (Equation 3) can still only equal unity, while in the actual case a value greater than unity is found.

While no simple correlation between heat flow and moisture movement is evident, the magnitude of the increase in heat flow observed in the tests with moisture does indicate some of the factors involved.

The larger of the two moisture contents in test No. 2 is seen to produce a much larger increase in heat flow.

Comparing tests Nos. 1 and 2, with equal total amounts of moisture, a much greater increase in heat flow resulted in test No. 2. This can be attributed to the lower thermal resistance between the heat source and the moisture source and also between the moisture sink and the heat sink.

The expanded vermiculite (test No. 3) exhibited a relatively smaller increase in heat flow. It is thought that the water present was more intimately bound in this material and permitted a much lower rate of evaporation.

In test No. 4 the higher temperature level test produced a larger increase in

heat flow. The 9 percent moisture content at 41 F indicates a relative humidity of 47.5 percent or a vapor pressure of 0.122 in. Hg. Upon increasing the temperature at $x = 0$ to 79 F the vapor pressure could rise momentarily to approximately 0.500 in. Hg, producing a vapor pressure difference of 0.378 in. Hg. The corresponding vapor pressure difference produced by increasing T_1 from 12 F to 47 F would only be 0.124 in. Hg, resulting in the smaller increase in heat flow. Tests at a 20 F temperature difference ($T_1 = 60$, $T_2 = 40$ not shown here) produced practically the same relative heat flow as a 40 deg temperature difference ($T_1 = 80$, $T_2 = 40$). With one-half the temperature difference, very nearly one-half the initial vapor pressure difference results and hence the same relative heat flows.

Since a vapor flow is necessary to produce the increased heat flows illustrated, the vapor permeability of the materials can be expected to have a marked influence on the increase in heat flow.

The transient conditions selected for study were chosen to facilitate the testing. They do, however, give some indication of the probable importance of moisture in cases of periodic heat flow such as are involved in the determination of air-conditioning loads from walls and roofs. In these studies the temperature changes were created suddenly while in periodic heat flow they occur gradually. The rapid and substantial increase in heat transmission attributable to the presence of moisture in certain of the transient cases indicates that similar substantial effects may be expected in periodic flow. The maximum periodic flow, the total flow or both may be increased. Since the moisture may be regarded as providing a mechanism which can, by providing for latent heat transfer, by-pass part of the resistance to conduction provided by a vapor permeable insulation, it may be expected also that in certain cases the time lag for maximum heat flow will be substantially decreased.

The cases studied were for closed systems, whereas in walls moisture can enter and leave at the boundaries. However, the amounts of moisture introduced in these tests of 0.032 and 0.065 lb per sq ft are relatively small and well within the amounts which might be expected in any hygroscopic materials regardless of moisture transfer at the boundaries. Built-up roofs will usually provide a close approximation to a closed system and may at times contain considerably greater quantities of water than those studied.

CONCLUSION

It has been shown that moderate amounts of moisture present in porous samples capable of providing moisture storage can modify greatly the transient heat flow conditions from those found for corresponding dry samples. Temperature changes produced corresponding vapor pressure changes and led to evaporation, vapor flow, and condensation. This constituted an additional mechanism for the transmission of heat in the form of latent heat, operating in conjunction with the usual mechanisms of heat transmission in dry materials. The effect of the moisture upon the steady state heat transmission in the samples studied was shown to be small.

The magnitude of the variation in the heat flow pattern produced by the presence of moisture in an actual wall or roof construction would appear to depend on: (*a*) the amount of moisture present; (*b*) the imposed temperatures and vapor pressures; (*c*) the moisture-sorbing and moisture transmission properties of the materials; and (*d*) the heat transmission properties of the materials and the arrangement of the materials.

The results indicate that substantial effects may also be found in many cases of periodic heat flow in walls and roofs. Further experimental work should be undertaken to determine how extensive these effects may be in practice.

ACKNOWLEDGMENT

The experimental work was done by the author while on the staff of the Prairie Regional Station of the Division of Building Research, National Research Council, at Saskatoon, Canada. It formed the basis for a thesis submitted, under a co-operative arrangement, to the Department of Mechanical Engineering, University of Saskatchewan, in partial fulfillment of the requirements for the degree of Master of Science. The author is indebted to members of the staff of the College of Engineering of the University of Saskatchewan, to N. B. Hutcheon, assistant director of the Division of Building Research, and to G. O. Handegord, officer-in-charge of the Prairie Regional Station, for assistance and encouragement in the prosecution of this project.

This paper is a contribution from the Division of Building Research, National Research Council, Canada, and is published with the approval of the director.

DISCUSSION

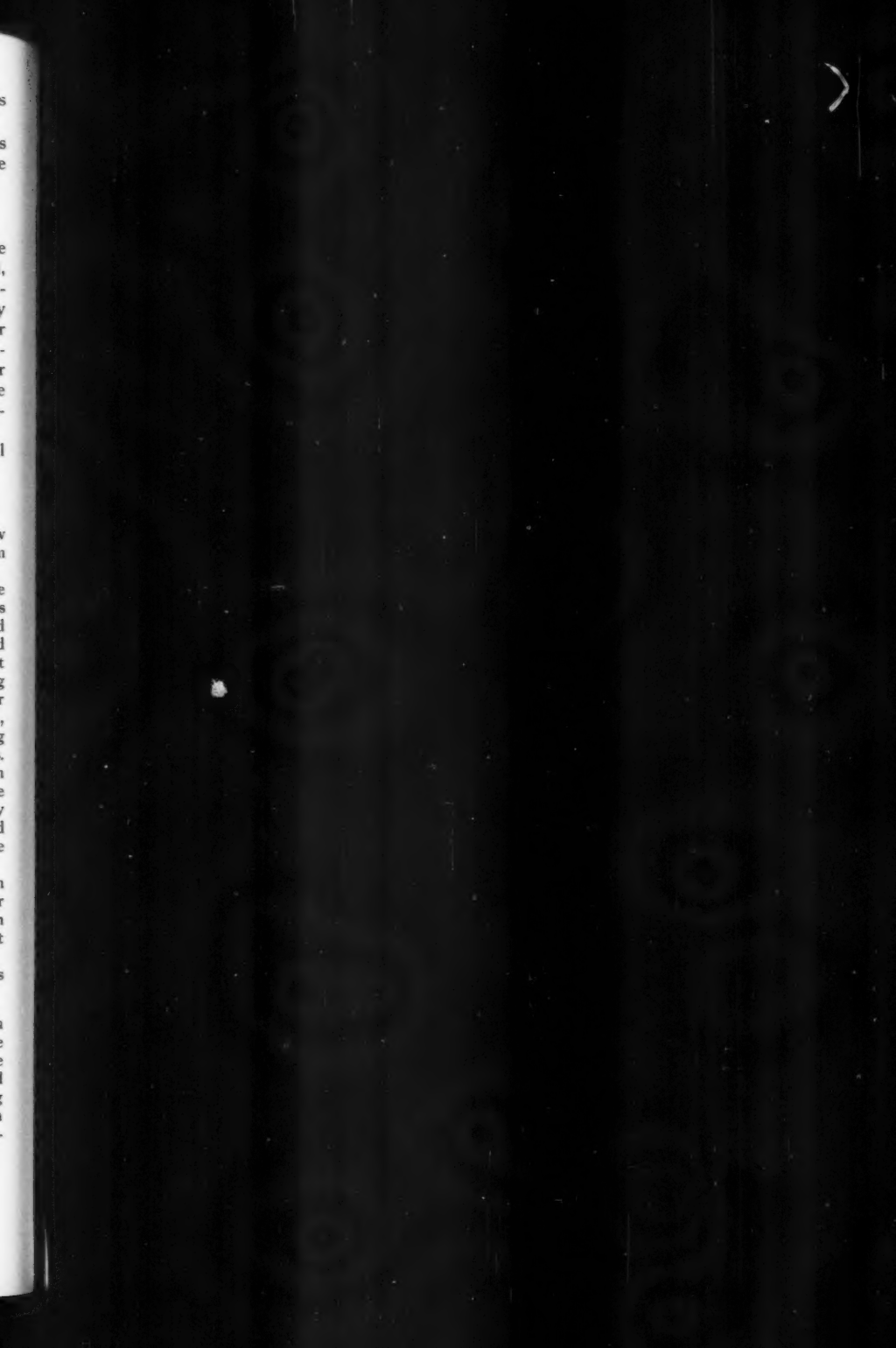
B. L. PALMER, Jackson, Miss.: The discussion is not exactly confined to the flow in the building material but an experience that I had in the past year in which a gypsum board roof was used on a building in which I put air-conditioning equipment.

The roof was formed by using what are known as bulb tees. Amounts to a miniature I-beam with a top flange cut off and a bulb substituted. At the bottom fiber glass insulating board was placed as a form board, then gypsum was poured on the top and the roof was laid right on this gypsum as a deck. It was put on in early spring and was quite wet. In fact there was a good deal of rain at the time. It was kept wet until it was covered by the roofer but when the sun came out we had the very interesting experience of those bulb tees which protruded up through the insulating board fiber glass and it was practically in contact. Well, it went through part of the gypsum, almost up to the top of the room. The gypsum was several inches thick; the insulating board 1 in. thick; the bottom of the bulb tees dripped and damaged floors and ceilings. This meant trouble for the contractors. Moisture migration, as described so well in this paper, was taking place. The sun was drying out the wet gypsum deck and the moisture was migrating into the building, and condensing on the bulb tees in the early morning. The wet roof temperatures were low and the bulb tees readily conducted temperature of the cold roof down to the bottom side of the slab and the high moisture of the air would lower it, built up from the day before, causing the condensation.

The remedy affected in this building was to cut some holes in the duct-way that ran above the ceiling between the ceiling and the roof and just let some conditioned air spill out into that space and purge out the high vapor content of the air trapped between the ceiling and the roof. This went on until the weather got hot enough to dry out the gypsum slab.

I haven't seen any condensation this winter. I went back in some of those buildings and looked to see what happened to that flow if that gypsum ever really dried out.

AUTHOR'S CLOSURE: The situation described by Mr. Palmer illustrates clearly an extreme case of moisture movement under reversing temperature gradients. The moisture present in this example was sufficient to cause severe condensation and the formation of free water. Even without free water present the same movement will occur with moderate amounts of moisture in porous materials. The accompanying latent heat exchange is perhaps of great significance in the calculation of maximum heat loads and the lag between maximum heat load and maximum temperature difference.



1562

No. 1562

CHARACTERISTICS OF DOWNWARD JETS FROM A VERTICAL DISCHARGE UNIT HEATER

By S. M. YEN*, LINN HELANDER**, AND L. B. KNEE†, MANHATTAN, KANS.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with the Engineering Experiment Station of Kansas State College.

UNDER the guidance of the TAC on Air Distribution, an exploratory study of jets of heated air from a vertical discharge unit heater was undertaken as an intermediate step in the jointly sponsored research on downward jets of heated air being conducted at Kansas State College. The objectives of this exploratory study were to: (1) Obtain data descriptive of the behavior of vertical jets of heated air of the type encountered in practice; (2) Compare and correlate data on the down-throw of such jets with data on the down-throw of jets from long radius *ASME* nozzles published in a previous paper[1]; (3) Obtain guidance for future research.

The second of these objectives was dealt with in a paper[2] presented before the Society at its Semi-Annual meeting in June 1954. That paper gave data on the down-throw of heated jets from a vertical discharge unit heater for 3 types of outlets and by means of a throw factor, f, correlated these data with corresponding data on jets discharged from long-radius *ASME* nozzles. In addition, the paper gave data on the pattern of flow over a distance approximately 3.5 outlet diameters down from the outlet.

The present paper deals with: (1) a study of the remaining portion of the jet discharged from the unit heater, that is, with the section that extends downward from a level approximately 2 outlet diameters below the outlet; and (2) with the distribution of temperatures in the space surrounding the jet. Two of the 3 outlets previously employed were used: the annular outlet and the shallow diffuser.

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* Assistant Professor, Department of Mechanical Engineering, Kansas State College.

** Professor and Head, Department of Mechanical Engineering, Kansas State College. Member of ASHAE.

† Instructor, Department of Mechanical Engineering, Kansas State College.

[1] Exponent numerals refer to References.

Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

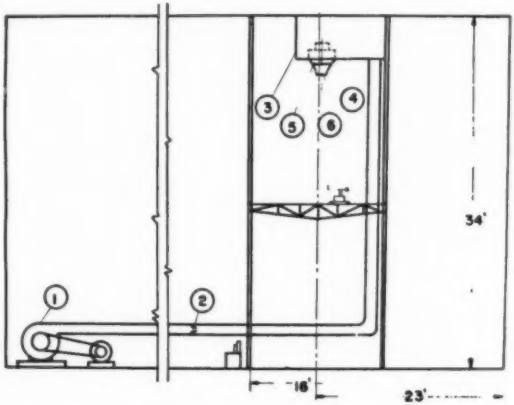

FIG. 1—SCHEMATIC ARRANGEMENT OF EQUIPMENT. REFERENCE NUMBERS: (1) BLOWER; (2) MEASURING NOZZLE; (3) PLENUM CHAMBER; (4) HEATER; (5) FAN; AND (6) OUTLET

Two non-isothermal jets and 1 isothermal jet discharged from the annular outlet and 1 non-isothermal jet discharged from the shallow diffuser were investigated. The data obtained are presented graphically:

1. As plots of velocities and temperatures along the vertical axis of the jet discharged from the unit heater;

2. As plots of velocities and temperatures along the diameter of the jet at representative levels;

3. As maps of constant temperature contour lines indicative of the distribution of temperatures in the space surrounding the jet.

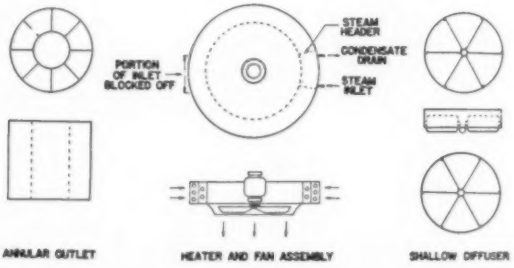

FIG. 2—SCHEMATIC DRAWING OF HEATER AND OUTLETS. FOR SYMMETRY; A SEGMENT OF THE AIR INLET EQUIVALENT TO THE STEAM HEADER WAS BLOCKED OFF

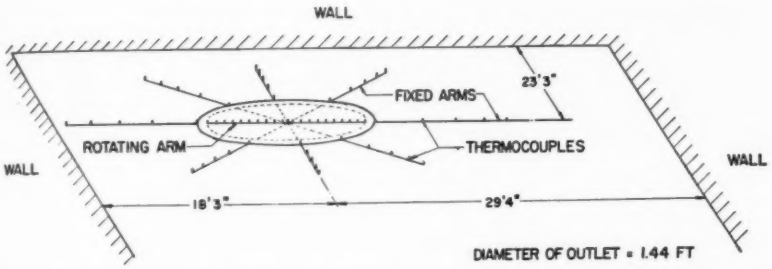

FIG. 3—AN ASSEMBLY OF FIXED, RADIAL ARMS, EACH WITH FIVE THERMOCOUPLES, AND ONE ROTATING CENTRAL ARM WITH 21 THERMOCOUPLES WERE USED TO OBTAIN THE HORIZONTAL DISTRIBUTION OF TEMPERATURE IN THE SPACE SURROUNDING THE JET. THE ASSEMBLY WAS MOVED VERTICALLY TO REPRESENTATIVE LEVELS. AT EACH LEVEL 121 TEMPERATURE READINGS WERE OBTAINED

Simple formulas are given for the velocities and the temperatures along the vertical axis of the jet.

Distribution of velocities and temperatures in the heated jet did not support the use of simplifying assumptions which imply similarity in the pattern of flow at different levels of a vertical jet of heated air. However, velocities and temperatures along the axis of the jet were not inconsistent with analytically developed equations based on such simplifying assumptions. In the space surrounding the jet, the constant-temperature contour lines close to the jet followed the form of the jet. Remote from the jet these lines were asymptotic to horizontal lines.

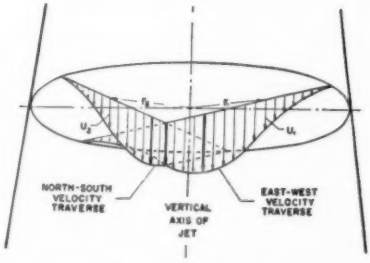

FIG. 4—VELOCITIES WERE MEASURED ALONG NORTH-SOUTH AND EAST-WEST CHORDS WHICH INTERSECTED ON THE VERTICAL AXIS OF THE OUTLET. THESE VELOCITIES WERE REPLOTTED AGAINST THEIR DISTANCE, r, FROM THE JET AXIS. THUS U_1 WAS PLOTTED AGAINST r_1, U_2 AGAINST r_2, ETC., TO FORM PROFILES BASED UPON THE JET DIAMETER

The estimated apparent boundary forces, evaluated as unaccounted-for forces acting on the jet, were found to be substantial. The hourly rate of room air entrainment per foot of travel was found to be a constant for the isothermal jet over the range of travel investigated. For the non-isothermal jets, it decreased with travel and became zero at, roughly, the mid-point of travel, after which the mass rate of flow progressively decreased.

EQUIPMENT AND PROCEDURE

Fig. 1 shows the test set-up. The unit heater was housed in an elevated plenum chamber in which atmospheric pressure was maintained by means of an external

TABLE 1—TEST CONDITIONS

	ANNULAR OUTLET			SHALLOW DIFFUSER
	NON-ISOTHERMAL		ISOTHERMAL	NON-ISOTHERMAL
	CASE I	CASE II		
	1	2	3	4
Gage Pressure in Plenum Chamber	0	0	0	0
Steam Pressure, psig	25	50	..	50
Outlet diameter, D_o, ft	1.44	1.44	1.44	1.44
Flow at outlet, cfm	1620	1610	1640	1440[a]
Avg Outlet Vel., $\bar{U}_o$, fps	16.6	16.5	16.8	14.7
Avg Outlet Vel., fpm	994	987	1000	882
Heat De ivery, Btu/hr	56,900	86,800	..	80,100
Avg Out et Temp., $\bar{T}_o$, R deg	584	597	..	596
Avg Outlet Temp., F deg	124	137	..	136
Room Temp., T_a, R deg	543	537	544	535
Room Temp., F deg	83	77	84	75
Avg Temp. of Air entering Plenum Chamber	86	79	..	76
Relative Outlet Downthrust	1.35	1.45	..	1.28
$\beta_o = [\bar{U}_o^2/(gD_o)]/(\bar{T}_o/T_a - 1)$	78	52	..	41
Throw, L_{max}, feet	22.5	18.5	..	12.5
Throw Factor, $f = (L_{max}/D_o)/(1.7\sqrt{\beta_o})$	1.04	1.05	..	0.80

[a] Adjustable guide vanes arbitrarily set vertical, not in position for maximum output

blower. Fig. 2 shows the unit heater and the 2 outlets employed. The annular outlet had vertical guide vanes; the shallow diffuser had adjustable guide vanes set in their vertical position. No attempt was made to determine the most favorable setting of the vanes for the shallow diffuser, and the vertical setting employed may not have been a favorable setting from the point of view of operating performance, although satisfactory for the purposes of the tests.

The techniques and procedures previously used and described were again employed. The bottom of the jet was located in each case by simultaneously moving a thermocouple and a Prandtl tube upward into the stream and noting the first point at which both gave evidence of being in the jet. This point was approxi-

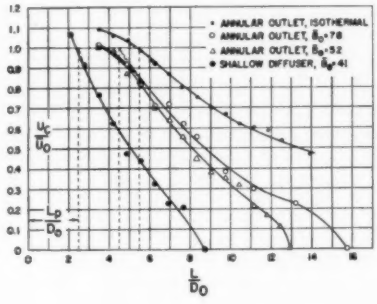

FIG. 5—MAXIMUM LOCAL VELOCITY, U_c, DEFINED AS VELOCITY ON VERTICAL AXIS OF JET. L/D_o = DISTANCE DOWN FROM FACE OF OUTLET IN OUTLET DIAMETERS. $\bar{U}_o$ = AVERAGE VELOCITY AT OUTLET. VALUES PLOTTED ARE VALUES OBTAINED BELOW THE LEVEL AT WHICH FULLY DEVELOPED FLOW WAS FIRST ESTABLISHED

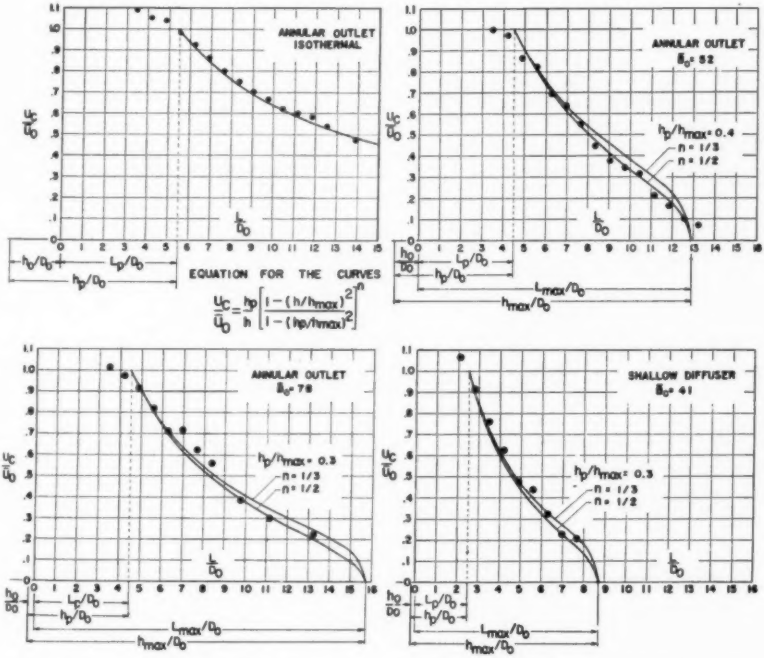

FIG. 6—DATA FROM FIG. 5 REPLOTTED TO SHOW ORDER OF AGREEMENT WITH EQUATION 1, REPRESENTED BY SOLID LINES. h_o AND n WERE DETERMINED BY A CUT AND TRIAL MATCHING OF THE EQUATION WITH TEST DATA. U_c = VELOCITY AT AXIS OF JET, $\bar{U}_o$ = VELOCITY AT OUTLET, L = DISTANCE DOWN FROM OUTLET, AND D_o = OUTLET DIAMETER

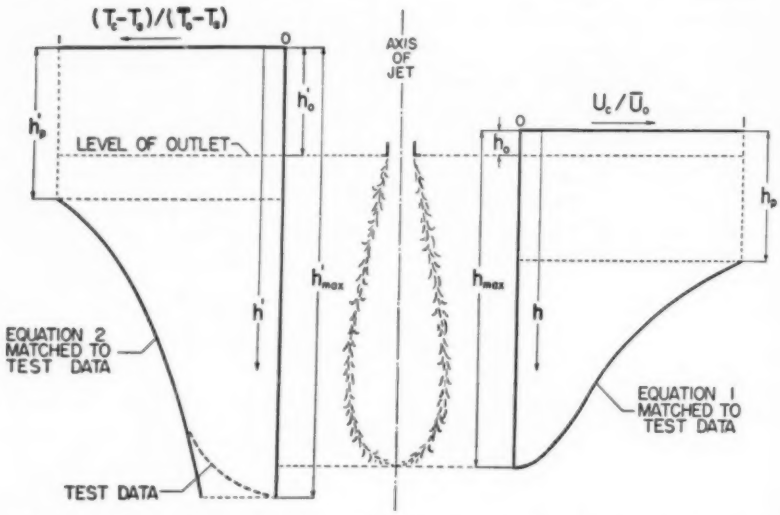

FIG. 7—SCHEMATIC DRAWING ILLUSTRATING SIGNIFICANCE OF SYMBOLS USED IN EQUATIONS 1 AND 2. h'_o, WHICH LOCATES THE ORIGIN, O, FOR PLOTTING $(T_c - T_a) / (\overline{T}_o - T_a)$ *vs.* h', AND h_o, WHICH LOCATES THE ORIGIN FOR PLOTTING $U_c/\overline{U}_o$ *vs.* h, WERE DETERMINED BY A TRIAL AND ERROR MATCHING OF EXPERIMENTAL DATA AND THE EQUATIONS. THE SUBSCRIPT p INDICATES THE POINT AT WHICH THE ORDINATE $(T_c - T_a) / (\overline{T}_o - T_a)$ OR $U_c/\overline{U}_o$ HAS A VALUE OF UNITY

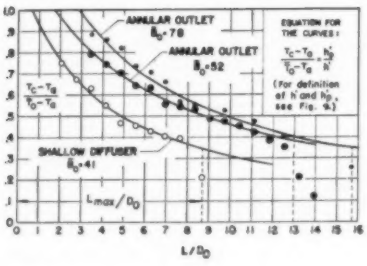

FIG. 8—RATIO OF TEMPERATURE ELEVATION, $(T_c - T_a)$, ALONG AXIS OF JET FOLLOWING THE ESTABLISHMENT OF FULLY DEVELOPED FLOW TO TEMPERATURE ELEVATION AT OUTLET, $(\overline{T}_o - T_a)$, DISTANCE FROM OUTLET IN OUTLET DIAMETERS. THE SOLID LINES REPRESENT EQUATION 2. AGREEMENT WITH THE EQUATION IS SATISFACTORY EXCEPT FOR THE POINTS THAT FOLLOW THE BREAK IN CURVE OF PLOTTED DATA NEAR THE BOTTOM OF THE JET

mately 1 outlet diameter above the point which would have been established as the bottom of the jet by temperature readings alone. Temperatures and velocities measured within the boundaries of the jet were recorded by means of single point, high-speed, electronic, curve-drawing instruments. Temperatures in the space surrounding the jet were recorded by means of a 20-point, high-speed, electronic, curve-printing recorder. This unit was connected to a four-bank external switch.

To establish outlet conditions, comprehensive velocity and temperature traverses were made at the face of the outlet. Maximum local velocities and temperatures in the region of fully developed flow were located and plotted by carefully exploring the central region of the jet at levels approximately 1 ft apart.

Fig. 3 shows the arrangement and locations of the thermocouples installed on the instrument carriage for measuring temperatures in the space surrounding the jet. A total of 121 readings were taken at each level explored, beginning with a level located 2 outlet diameters below the outlet face and ending with a level approximately 4 outlet diameters above the floor.

The mass rate of flow and the momentum flux were evaluated from velocity and temperature profiles established by the procedure illustrated in Fig. 4.

TEST CONDITIONS

The test conditions under which the jets discharged from the unit heater were investigated are tabulated in Table 1. With the annular outlet, 3 different jets were investigated; 2 non-isothermal jets with different outlet temperatures and 1 isothermal jet. Only a non-isothermal jet was investigated using the shallow diffuser. In all tests, atmospheric pressure was maintained in the plenum chamber.

The test conditions employed permit one to compare the performances of: (1) an isothermal and a non-isothermal jet discharged from the same outlet; (2) two non-isothermal jets with different heat quantities discharged from the same outlet; and (3) two non-isothermal jets delivering approximately the same heat quantity discharged from 2 different outlets.

The temperature distribution in the space surrounding the jets was established by a separate investigation using both outlets. The test conditions were the same as those listed in Table 1, Columns 2 and 4, but the throws obtained were 8 percent greater than the throws tabulated in these columns.

In comparing the test data in Table 1 with corresponding data of the previous paper, it should be noted, that the figures in Table 1 resulted from tests with a fan, the blades of which had appropriate tip clearances. Two fans were used for the tests reported in the previous paper, but one of these had blades the tip clearances of which were excessive.

DETAILED EXPERIMENTAL DATA AND RESULTS

Although broad generalizations cannot be drawn from the experimental data obtained, various significant characteristics have been revealed. These are presented in the following as being of both practical and theoretical interest:

1. Velocities and Temperatures along Vertical Axis of Jets: Fig. 5 is a graph of velocities along the vertical axis of the jets investigated (defined as the locus of maximum local values) plotted against distance from the outlet in outlet diameters. The ordinate U_c/U_o is the ratio of the velocity at the axis to the average velocity

at the outlet evaluated from the volume rate of flow. In Fig. 6, velocities plotted in Fig. 5 are replotted to show the order of agreement with Equation 1. This equation is a correlating equation the form of which was obtained analytically on the basis of simplifying assumptions. It represents the ratio of the centerline velocity, U_c, to the average velocity, $\overline{U}_o$, as a function of 3 factors (refer to Fig. 7):

(*a*) h_p, the distance from an origin on the vertical axis of the jet h_o ft above the face of the outlet to the point on the axis where U_c/U_o from Equation 1 has a value of unity;
(*b*) h, the distance from the same origin to the point under consideration on the axis; and
(*c*) h_{max}, the distance from the origin to the bottom of the jet.

The exponent, n, in Equation 1, as well as h_o is treated as an empirical factor. Values of these factors were obtained by a cut and trial matching of data with curves representing the equation:

$$\frac{U_c}{\overline{U}_o} = \frac{h_p}{h}\left(\frac{1-\left(\frac{h}{h_{max}}\right)^2}{1-\left(\frac{h_p}{h_{max}}\right)^2}\right)^n \quad \ldots\ldots\ldots\ldots (1)$$

Equation 1 with n = 1/3 is equivalent to Koestel's equation[3] for centerline velocities. With n between 1/3 and 1/2, the equation matches the experimental data well. However, undue significance should not be attached to this apparent matching of test data with theory. In the first place, the exponent, n, is sensitive to the terminal shape of the curve of axial velocities *vs.* travel, and this shape must, in some measure, be estimated. In the second place, the derivation of the equation was based on convenient, simplifying assumptions.

Fig. 8 shows temperatures along the vertical axis of the jets plotted against distance down from the outlet in outlet diameters. The solid curves represent the equation:

$$\frac{T_c - T_a}{\overline{T}_o - T_a} = \frac{h'_p}{L + h'_o} = \frac{h'_p}{h'} \quad \ldots\ldots\ldots\ldots (2)$$

The left-hand side of this equation is the ratio of the temperature elevation along the vertical axis of the jet to the temperature elevation at the outlet. The right-hand side is simply the product of h'_p (see Fig. 7) and the reciprocal of the quantity obtained by adding h'_o to the travel, L. The manner in which h'_p and h'_o were evaluated is illustrated in Fig. 9.

Table 2 gives values of h_o/D_o, h'_o/D_o, h_p/D_o and h'_p/D_o.

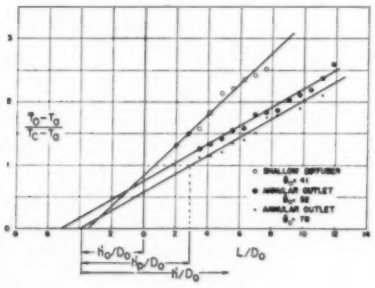

FIG. 9—IN THIS FIGURE, THE RECIPROCAL OF THE ORDINATES OF FIG. 8 HAS BEEN PLOTTED AGAINST TRAVEL, L/D_o. IN AGREEMENT WITH EQUATION 2, THE POINTS PLOTTED ARE REPRESENTED BY STRAIGHT LINES. L/D_o IS MEASURED FROM THE ZERO POINT; h'/D_o FROM A POINT h'_oD_o OUTLET DIAMETERS TO THE LEFT OF THE ZERO POINT; h'_p/D_o IS THE VALUE OF h'/D_o WHERE $(T_c - T_a) / (T_o = T_a) = 1$

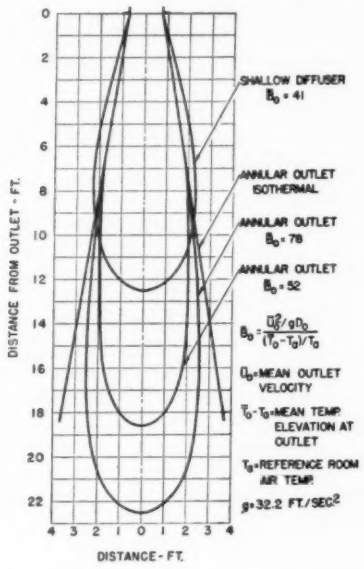

FIG. 10—APPROXIMATE BOUNDARIES OF JETS DETERMINED FROM VELOCITY PROFILES. DIAMETER OF OUTLET = 1.44 FT. FOR OUTLET CONDITIONS, REFER TO TABLE 1

2. Jet Boundaries: The approximate boundaries of the jets studied are shown in Fig. 10. These were established by means of Prandtl tube determinations of the surface of zero downward velocity. This method provides comparable, qualitative data on the jet envelopes, but not definitive data.

3. Temperature Distribution in the Space Surrounding the Jets: Figs. 11 and 12 show the distribution of temperatures in the space surrounding each of 2 jets, one discharged from the annular outlet, the other from the shallow diffuser. The lines

TABLE 2—VALUES OF h_o/D_o, h'_o/D_o, h_p/D_o AND h'_p/D_o
(See Fig 7, also Figs 6 and 9)

	h_o/D_o	h'_o/D_o	h_p/D_o	h'_p/D_o
Annular Outlet				
(1) Isothermal Jet	2.5	..	8.6	..
(2) $B_o = 78$	0.28	4.00	4.80	6.90
(3) $B_o = 52$	1.11	5.20	5.63	6.95
Shallow Diffuser				
$B_o = 41$	0.15	3.50	2.65	4.20

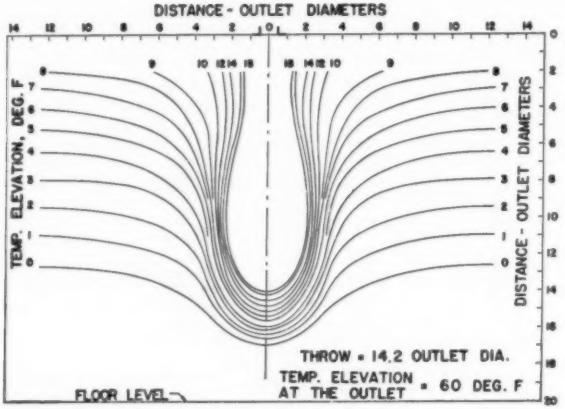
DISTANCE - OUTLET DIAMETERS
TEMP. ELEVATION, DEG. F
DISTANCE - OUTLET DIAMETERS
THROW = 14.2 OUTLET DIA.
TEMP. ELEVATION AT THE OUTLET = 60 DEG. F
FLOOR LEVEL

FIG. 11—DISTRIBUTION OF TEMPERATURE IN SPACE SURROUNDING THE JET FROM ANNULAR OUTLET. CONTOUR LINES OF CONSTANT-TEMPERATURE ELEVATION ABOVE REFERENCE ROOM TEMPERATURE MEASURED 30 IN. ABOVE THE FLOOR AT A POINT REMOTE FROM THE JET

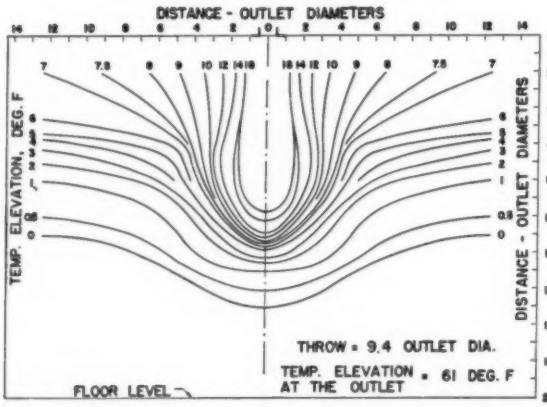
DISTANCE - OUTLET DIAMETERS
TEMP. ELEVATION, DEG. F
DISTANCE - OUTLET DIAMETERS
THROW = 9.4 OUTLET DIA.
TEMP. ELEVATION AT THE OUTLET = 61 DEG. F
FLOOR LEVEL

FIG. 12—DISTRIBUTION OF TEMPERATURE IN SPACE SURROUNDING THE JET FROM SHALLOW DIFFUSER. CONTOUR LINES ARE LINES OF CONSTANT-TEMPERATURE ELEVATION ABOVE REFERENCE ROOM TEMPERATURE MEASURED 30 IN. ABOVE THE FLOOR AT A POINT REMOTE FROM THE JET

drawn on these figures are lines of constant elevation of temperature, $(T - T_a)$. Each point used to establish these lines is the average of 8 temperature readings distributed in a circle about the vertical axis of the jet employed.

In the immediate vicinity of the jet, the lines of constant temperature elevation followed the contour of the jet. Remote from the jet, they became asymptotic to a horizontal line. At the immediate boundary of the jet, the temperature elevation decreased rapidly in a radially outward direction. In the space remote from the jet, a substantial, vertical temperature gradient existed. Immediately below the jet, there existed a region of elevated temperature. In this region, an intermittent upward flow of air occurred, induced by the elevated temperature.

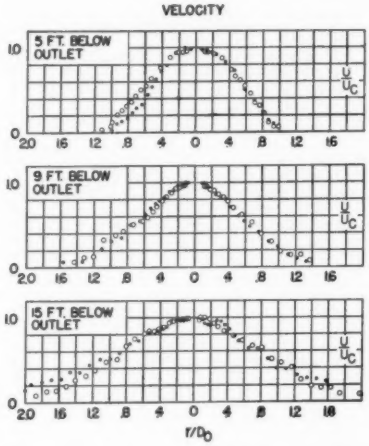

FIG. 13—VELOCITY PROFILES OF ISOTHERMAL JET FROM ANNULAR OUTLET. THE ORDINATE, U/U_c, IS THE RATIO OF THE LOCAL VELOCITY, U, AT DISTANCE r/D_0 OUTLET DIAMETERS FROM THE JET AXIS TO THE VELOCITY, U_c, AT THE AXIS

The temperature distribution represented by Figs. 11 and 12 should not be regarded as representative of practice, since the unit heater was employed, not as it ordinarily would be in practice, but to produce a jet that could be regarded as being an approximately free jet. The following factors, in this regard, are significant:

(a). The bottom of the jet from the annular outlet was 8.5 ft, that of the jet from the shallow diffuser 15 ft, above the level of the floor.

(b). There was no ceiling at the level of the outlet.

(c). The base room temperature, T_a, was measured at a level 30 in. above the floor at a point remote from the jet.

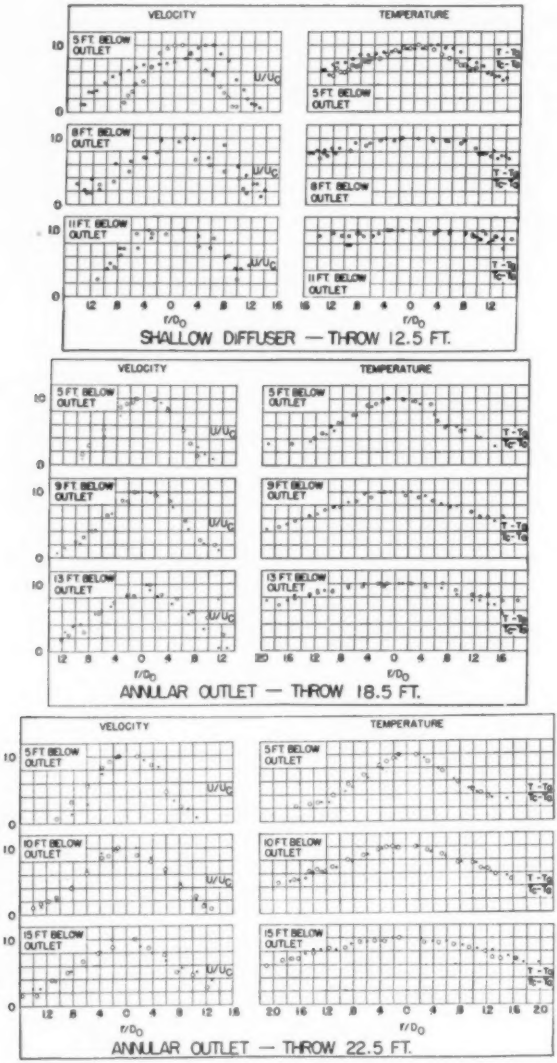

FIG. 14—VELOCITY PROFILES, LEFT, AND TEMPERATURE PROFILES, RIGHT, OF JETS FROM ANNULAR OUTLET AND SHALLOW DIFFUSER. U/U_c = RATIO OF LOCAL VELOCITY, U, AT DISTANCE r/D_o OUTLET DIAMETERS FROM AXIS TO VELOCITY, U_c, AT AXIS. $(T - T_a) / (T_c - T_a)$ = RATIO OF LOCAL TEMPERATURE ELEVATION, $T - T_a$, TO TEMPERATURE ELEVATION, $T_c - T_a$, AT AXIS

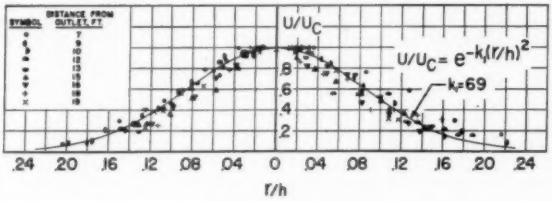

FIG. 15—COMPOSITE PLOT OF ALL VELOCITY PROFILES FOR JET FROM ANNULAR OUTLET WITH $\overline{B}_o = 78$ AND THROW = 22.5 FT. (SEE FIG. 14 FOR PROFILES AT REPRESENTATIVE LEVELS). THE SPREAD OF POINTS INDICATES THAT SIMILARITY IN THE PATTERN OF FLOW WAS NOT MAINTAINED ALONG THE PATH OF TRAVEL. r = RADIAL DISTANCE FROM AXIS OF JET; h = DISTANCE DOWN FROM ORIGIN h_o/D_o OUTLET DIAMETERS ABOVE FACE OF OUTLET. $h_o/D_o = 0.28$

(d). In the absence of artificial heating, a temperature difference of 3 deg existed between the temperature of the room at the level of the outlet and the temperature 30 in. above the floor.

4. Cross Jet Velocity and Temperature Profiles: Figs. 13 and 14 show representative velocity and temperature profiles of the type employed for evaluating the mass rate of flow, the momentum flux, and the buoyancy forces acting on the jets. These velocity and temperature profiles were obtained by superposing, in one case, profiles of velocities, in the other, profiles of temperatures, obtained along 2 diameters at right angles to each other. In Fig. 15, the velocity profiles obtained at representative levels of one of the jets have been replotted against a common dimensionless coordinate to form a single mean curve. This mean curve is satisfactorily represented by the error equation given in the figure with K_1 in that equation set equal to 69. However, extreme values of K_1 for the band of data plotted have a ratio of approximately 2 to 1.

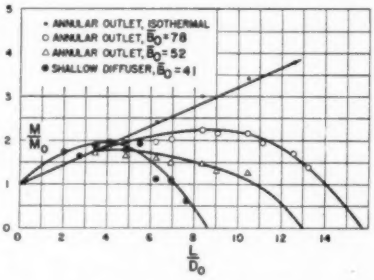

FIG. 16—RATIO OF MASS RATE OF FLOW, M, AT LEVEL L/D_o OUTLET DIAMETERS DOWN FROM OUTLET TO MASS RATE OF FLOW, M_o, AT OUTLET

5. Flow Rate and Air Entrainment: Fig. 16 shows the mass rate of flow plotted against travel in outlet diameters. The curve for the isothermal jet is a straight line. This signifies that the amount of room air entrained each hour over each foot of travel was essentially a constant for the extent of the jet investigated. The curves for the non-isothermal jets are concave viewed from the x axis. This indicates that the non-isothermal jets entrained air at a decreasing rate over the first part of their travel and then ejected air to the surrounding space.

6. Buoyancy Force and Jet Momentum: The difference between the change in the momentum flux and the buoyancy force is the sum of the apparent boundary forces, or unaccounted-for forces. Fig. 17 is a graph of such forces expressed as a fraction of the momentum flux at the outlet. It is quite possible that the boundaries of the

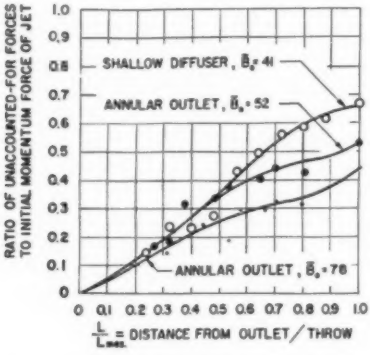

FIG. 17—UNACCOUNTED-FOR FORCES (APPARENT BOUNDARY FORCES) ACTING ON THE JET BETWEEN THE OUTLET AND LEVELS *L* FEET DOWN FROM THE OUTLET, EVALUATED AS THE DIFFERENCE BETWEEN THE DECREMENT OF MOMENTUM FLUX AND THE IMPRESSED BUOYANCY FORCE

jets were not ascertained with an accuracy sufficient to evaluate the buoyancy forces, and that this accounts to some extent for the magnitude of the unaccounted-for forces. However, unaccounted-for forces of the same order of magnitude were obtained by Nottage[4] in his experiments on a chilled jet.

EXAMPLE ILLUSTRATING USE OF DATA PRESENTED

Problem: Determine the temperature and velocity at the face of the outlet for a jet of heated air projected downward from a unit heater so as to reach the floor level with a maximum local velocity of 120 fpm and a maximum local temperature elevation of 15 deg above a reference room temperature of 70 F. The diameter of the outlet is 1 ft and the distance from the face of the outlet to the floor is 20 ft. The throw factor, *f*, of the outlet is assumed to be 0.90.

In terms of the symbols in the Nomenclature the following are the data given:

D_o = diameter of outlet = 1 ft.
f = throw factor = 0.90.
t_a = reference room temperature = 70 F.
T_a = absolute reference room temperature = 530 R (Rankine degrees).
L = distance from the outlet to the level of the point under consideration = 20 ft.
U_c = velocity at axis of jet L ft down from outlet = 120/60 = 2 fps.
t_c = temperature at axis of jet L ft down from outlet = 15 + 70 = 85 F.
T_c = absolute temperature at axis of jet L ft down from outlet = 545 R.

The problem is to find:

(1) $\overline{T}_o$ = mass mean temperature at outlet, Rankine degrees.
(2) $\bar{U}_o$ average velocity at outlet, feet per second.

Solution: From Equation 2:

$$\frac{T_c - T_a}{\overline{T}_o - T_a} = \frac{h'_p}{L + h'_o} = \frac{D_o}{D_o}\frac{h'_p}{L + h'_o} = \frac{\dfrac{h'_p}{D_o}}{\dfrac{L}{D_o} + \dfrac{h'_o}{D_o}} \quad \ldots\ldots (3)$$

where

h'_p and h'_o = empirical factors (see Table 2 for values of h'_p/D_o and h'_o/D_o).

Let $h'_p/D_o = 5$; $h'_o/D_o = 4$
$\overline{T}_o - T_a = (T_c - T_a)[(L/D_o + h'_o/D_o)/h'_p/D_o] = 15\,[(20 + 4)/5] = 72$
$\overline{T}_o = T_a + 72 = 530 + 72 = 602$ R
$\bar{t}_o = 142$ F

Now apply Equation 1:

$$\frac{U_o}{\bar{U}_o} = \frac{h_p}{h}\left(\frac{1 - \left(\dfrac{h}{h_{max}}\right)^2}{1 - \left(\dfrac{h_p}{h_{max}}\right)^2}\right)^n$$

where

$h = L + h_o$
$h_{max} = L_{max} + h_o$
h_p and h_o = empirical factors (see Table 2 for values of h_o/D_o and h_p/D_o).

Let $n = \frac{1}{2}$; $h_o/D_o = 0.5$; $h_p/D_o = 4$.

Since $U_c = 2$ fps at $L = 20$ ft,

$$\frac{2}{\bar{U}_o} = \left(\frac{4}{20+0.5}\right)\left[\frac{1-\left(\dfrac{20+0.5}{L_{max}+0.5}\right)^2}{1-\left(\dfrac{4}{L_{max}+0.5}\right)^2}\right]^{\frac{1}{2}}$$

$$= \left(\frac{4}{20.5}\right)\left[\frac{(L_{max}+0.5)^2-(20.5)^2}{(L_{max}+0.5)^2-(4)^2}\right]^{\frac{1}{2}} \quad \text{-------(4)}$$

The throw, L_{max}, is given by the following equation:

$$\frac{L_{max}}{D_o} = 1.70 f \left(\frac{\frac{\bar{U}_o^2}{gD_o}}{\frac{\bar{T}_o}{T_a} - 1} \right)^{\frac{1}{2}} \quad \ldots \ldots \ldots \quad (5)$$

Substituting the following values into this equation:

$$D_o = 1 \text{ ft}, f = 0.9, T_a = 530 \text{ R}, \bar{T}_o = 602 \text{ R}$$

$$L_{max} = 1.70\ (0.9)\ (\bar{U}_o)\ \{[1/32.2]/[(602/530) - 1]\}^{\frac{1}{2}}$$

$$L_{max} = 0.732\ \bar{U}_o \quad \ldots \ldots \ldots \quad (6)$$

Combining Equations 4 and 6,

$$2/\bar{U}_o = 0.195\ \{[(0.732\ \bar{U}_o + 0.5)^2 - (20.5)^2]/[(0.732\ \bar{U}_o + 0.5)^2 - (4)^2]\}^{1/2}$$

A graphical or cut and trial solution gives:

$$\bar{U}_o = 29.2 \text{ fps.}$$

Summary of Results from Solution:

Estimated velocity at outlet	= 29.2 fps
	= 1750 fpm
Estimated average temperature at outlet	= 142 F
Design velocity 20 ft below outlet if the floor is not present (20 ft below outlet is actual location of floor)	= 120 fpm
Maximum local temperature at floor*	= 85 F

CONCLUSIONS AND RECOMMENDATIONS

1. This paper, together with the previous paper on the same subject, gives data describing the behavior of jets of heated air projected vertically downward from a unit-heater into space with remote boundaries. These data, used with regard for the design of equipment and the test conditions employed, may serve as guidance and reference data for designers and investigators interested in developing a rational procedure for predicting the behavior and the performance characteristics of jets of the type investigated.

2. Velocities along the vertical axis of the jets in the region of fully developed flow were satisfactorily represented by an equation of the following form with n between ½ and ⅓:

$$\frac{U_o}{\bar{U}_o} = \frac{h_p}{h} \left(\frac{1 - \left(\frac{h}{h_{max}}\right)^2}{1 - \left(\frac{h_p}{h_{max}}\right)^2} \right)^n$$

3. Temperatures along the vertical axis of the jets in the region of fully developed flow were satisfactorily represented by an equation of the following form:

$$\frac{T_e - T_a}{\bar{T}_o - T_a} = \frac{h'_p}{h'}$$

* Use of the estimated outlet conditions should result in a floor level temperature that will not exceed 85 F, but may be less than 85 F.

In the vicinity of the bottom of the jet this equation gives values of

$$(T_o - T_a)/(\overline{T}_o - T_a)$$

that are somewhat higher than experimentally determined values and it is not useful for evaluating temperatures below the point on the axis where the velocity is zero.

4. The thermal effect of the jets extended below the bottom of the jets a distance of approximately 1 outlet diameter. Lines of constant temperature elevation above room temperature, drawn in the vicinity of the jets, had the shape of the jets. These lines, remote from the jets, became asymptotic to a horizontal line.

5. Air was entrained over only roughly the first half of the heated air jets. Beyond the approximate mid-point of travel, the mass rate of flow of these jets decreased progressively to a value of zero at the end of the throw. In the case of the isothermal jet, the amount of air entrained per hour per foot of travel was very nearly a constant over the vertical distance investigated.

6. The momentum flux of the isothermal jet decreased slightly with travel. In the case of the heated jets, the decrease of downward momentum flux was substantially greater than the buoyancy force.

7. The exploratory study of jets from a vertical discharge unit-heater of the type used in practice has served the purpose of providing useful guidance data. However, future research should be devoted to jets wherein the factors to be investigated can be isolated and controlled.

8. Means are needed for measuring directly the momentum force of a jet issuing from an outlet. An experimental device for this purpose has been designed and constructed and is now in the process of being tested.

9. In an experimental investigation of jets, it is important that the entire field of influence of the jets be explored.

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2. ASHVE RESEARCH REPORT NO. 1511—Characteristics of Downward Jets of Heated Air from a Vertical Discharge Unit Heater, by Linn Helander, S. M. Yen and L. B. Knee (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 359).

3. ASHVE RESEARCH REPORT NO. 1512—Computing Temperatures and Velocities in Vertical Jets of Hot or Cold Air, by Alfred Koestel (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 385).

4. ASHVE RESEARCH REPORT NO. 1459—Exploration of a Chilled Jet, by H. B. Nottage, I. G. Slaby, and W. P. Gojza (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 357).

GLOSSARY OF TERMS

Relative downthrust = ratio of actual downthrust at face of issuing jet to downthrust that a flat velocity profile would have developed with the same flow and the same outlet diameter, D_o.

Bottom of jet = point at which a thermocouple and a Prandtl tube simultaneously moved upward along the axis of jet from floor level give first distinct evidence of entering the jet.

Boundary of jet = surface of zero downward velocity as approximated by a Prandtl tube.

NOMENCLATURE

$\bar{B}_o$ = cross-stream average buoyancy number at outlet = $(\bar{U}_o^2/gD_o)/[(\bar{T}_o - T_a)/T_a]$.
c = subscript indicates value on axis of jet, defined as locus of maximum local values of temperature or velocity.
D_o = outlet diameter, feet.
g = gravitational acceleration, 32.2 feet per (second) (second).
f = throw factor.
h = vertical coordinate for locating velocities, feet.
h_o = distance from origin of h to face of outlet, feet.
h_p = distance from origin of h to point at which U_c on curve of U_c versus L/D_o extended equals $\bar{U}_o$, feet.
h_{max} = distance from origin of h to bottom of jet (see glossary of terms), feet.
h' = vertical coordinate for locating temperature, feet.
h'_o = distance from origin of h' to face of outlet, feet.
h'_p = distance from origin of h' to point at which $(T_c - T_a)$ on curve of $(T_c - T_a)$ versus L/D_o extended equals $(\bar{T}_o - T_a)$, feet.
h'_{max} = distance from origin of h' to point on axis of jet where $(T_c - T_a)$ is zero, ft.
L = distance down from outlet, feet.
L_p = $h_p - h_o$, feet.
L_{max} = distance from outlet to bottom of jet = $h_{max} - h_o$, feet.
M = mass rate of flow at any designated level, pounds per second.
M_o = mass rate of flow at outlet, pounds per second.
o = subscript referring to outlet.
r = radial distance from axis of jet to point of jet under consideration, feet.
T = local temperature at any designated point of jet, Fahrenheit absolute.
$\bar{T}_o$ = mass-mean temperature at outlet, Fahrenheit absolute.
T_c = temperature along axis of jet defined as locus of maximum local values, Fahrenheit absolute.
T_a = temperature of undisturbed room air measured at instrument table 30 in. above floor, Fahrenheit absolute.
t_a = reference room temperature, Fahrenheit degrees.
t_c = temperature at axis of jet L ft down from outlet, Fahrenheit degrees.
U = local velocity at any designated point of jet, feet per second.
$\bar{U}_o$ = cross-stream average velocity at outlet = (volume rate of flow)/$[\pi D_o^2/4]$, feet per second.
U_c = velocity along axis of jet defined as locus of maximum values, feet per second.

DISCUSSION

R. M. CONNER, Cleveland, Ohio (WRITTEN): In the first place let me say that frankly I am very sympathetic to all ASHAE. research endeavors. As a matter of fact they are to me our most important activities and the Society's future growth, and general usefulness will depend more and more as the years go by on their number and the extent of accuracy with which they are accomplished.

Second, as perhaps all of you know, heating and air-conditioning are both becoming of tremendous importance to the gas business. While both are relatively new, their growth, particularly of the former, has simply been phenomenal. No doubt this fact is largely due to the wide-spread use of a relatively cheap fuel (natural gas) over a large part of the United States and Canada. In fact one of the best indices of the extremely rapid growth of the former has perhaps been that during the past 15 years the number of heating appliances tested for approval by our establishments, compared to all other types, has risen from practically nothing to now roughly 50%. Little wonder we of the gas business are interested in heating.

One important type of gas heating appliance is the so-called unit heater, employed to heat various types of commercial and industrial establishments, auditoriums and many other buildings.

All the gas unit heaters that I know of discharge the heated air horizontally, thereafter it is deflected downward by a series of louvers initially in an almost vertical plane. In-so-far as the heating of relatively small areas is concerned where the capacity of heater is at or near the required Btu for the cubical contents of the room involved I can't see that the actual shape of the heated air envelope from the heater would make too much difference. Where so-called spot heating is involved however, and at present this is usually the case, the path and shape of the heated air becomes very important. Another angle also to be considered is the fact that the heated air jet from a gas unit heater would probably average about 250 F.

Without too much effort or a great deal more research work it seems to me that relevant data could be extended to cover such units as I have discussed. It is possible that the somewhat higher temperatures involved might necessitate additional tests, and that the difference in the shape of the air pattern might also make this advisable. The end result obtained however, would not only be advantageous to our business but to all other trades using roughly the type of construction, temperatures, air velocities we employ.

Finally I think that the authors of this paper are to be congratulated not only upon the technical excellence with which resulting test data has been presented but also upon overcoming a seeming lack of interest at times on the part of our Committees in their project. It is hoped that this work at Kansas State will be enlarged and continued until all common types of unit-heaters will be satisfactorily covered.

H. H. YOUSOUFIAN, New York, N. Y. (WRITTEN): I am interested primarily in the qualitative aspects of the method of distributing heated air vertically downward to heat a space for comfort applications. The authors are making valuable contributions toward illuminating this aspect of the problem.

Based on the present tests, are they able to draw any conclusions yet regarding the advantages and disadvantages of using either the annular outlet or the shallow diffuser to provide an optimum distribution of temperatures over a large area in the so-called *zone of occupancy*?

If more research is planned for the future, will diffusers be tested which provide a marked conical air discharge pattern?

A. KOESTEL, Cleveland, Ohio, (WRITTEN): I would like to congratulate Professor Helander and his associates on a paper containing a wealth of experimental data and many fine correlations. I am happy to see that there is agreement between the equations employed in our paper (Reference 3) and those of Professor Helander.

The key to the derivation of accurate equations for predicting jet velocities and temperatures lies in the values tabulated h_o/D_o, h'/D_o, h_p/D_o, h_p'/D_o in Table 2. When once these values are known the jet behavior can be accurately defined by Equations 1 and 2. We should channel some of our experimental effort into finding out what factors determine the values of the zone lengths h_o/D_o, h_o'/D_o, h_p/D_o and h_p'/D_o. I believe that these zone lengths may be a function of the Reynolds number and possibly the Buoyancy number.

H. B. NOTTAGE, Encino, Calif., (WRITTEN): It is wonderfully encouraging to see this type of data come forth. With more of the same, analyzed in terms of force-motion principles to yield local stresses and heat-flow rates, at last we shall begin to gain a grasp upon the true physical phenomena in spatial air distribution. Being convinced of the fundamental *orderliness*, at least in a statistical sense, of these physical happenings, I would like to have the authors comment upon the following:

1. What is their picture of the happenings within and around the small volumes of air at representative positions throughout the influenced regions?
2. Why does the jet follow semi-empirical relationships to a certain extent, when, at the same time, the assumptions which were involved in deriving these relationships have been shown to be invalid?

L. R. PHILLIPS, Hartford, Conn.: I would like to commend the authors on their significant contribution to what we still call the art of air-conditioning. It is trite even to mention this but nevertheless they have made quite a contribution to bringing this one subject (which is particularly interesting to me, because I am engaged in research and development of diffusion of air into space) one step further to an exact science.

I have a question or two. One is in relation to the throw factors spoken of. They relate to the annular diffuser and the shallow diffuser, and I believe in the previous paper the authors referred to another type of diffuser. What I would like to know is what range of this throw factor might be expected and how the authors would suggest using the equations they give with an unknown diffuser, that is, one that they have not yet tested.

Another point, I note on the figures, is the high temperature of the air above the space feeding into the constant temperature lines in the graph. To what do the authors attribute this high temperature in the above part of the space? Perhaps it would be advantageous to show the contour lines complete throughout the entire enclosure to get some idea where this high temperature or stratification originates.

E. K. CAMPBELL, Kansas City, Mo.: Our experience has been, in connection with large, warm air heating systems, heating large spaces where the diffusion of the heat was one of the main problems to be solved, generation of the heat was comparatively simple, but in some types of buildings the distribution of the heat was a complicated problem.

We have gotten away from the jet or the throw theory pretty much in our theories of distribution of heat within space, depending on the gravity movement of the air under different temperatures. Regardless of how the warm air behaves within the jet, and regardless of whether it is a small jet or a big one, (and we dealt mostly in big ones) after the force of the jet is lost, the force which drove it into the room, the warm air begins to float. It will float clear across a room of any given size (which offers no obstructions) evenly distributed because it is a layer of even weight.

Then, as the colder air, or return air, is pulled off underneath, the warm air is let down and evenly distributed over the room. That works out quite perfectly and a complete study of the jets and the action within the force of the propulsion that puts the air into the room, I think, should be extended to include the action of the air after that force is lost and the gravity forces take over.

That principle was applied to the Butler Field House, some 25 or 30 years ago. The room was over 332 ft, 6 in. long, 206 ft wide and 90 ft in the center, with an arch roof and resulted in only 3 deg variation on any given level and only 12 deg gradiant in 90 ft high. The gravity forces must be taken into account as well as the action within the force of the jet.

Butler University, however, is not a job based on a vertical downward discharge jet. It is a horizontal discharge and not of jet proportions, being at low velocity so that the floating takes effect almost immediately. The difference between that and a downward jet is that the downward jets have to be of pretty high velocity in order to get them to go down very far. The downward jet, spreading as it goes down, mixes with the air in the lower layers, which of course are colder. If the velocity is high enough, the jet will reach clear to the floor and strike the floor. In either case, the downward jet creates another center towards which the air of the room will move. As the air of the room mixes with the air of the jet, the temperature of the mixture rises enough so that it floats to a higher level, not clear to the top, but to a higher level. So, the principles of distribution are really the same whether its a downward jet of high velocity or a horizontal jet of low velocity or anything between. The ultimate distribution of temperature within the room does not depend upon the force of the jet, it depends upon the total volume of warm air put in and the total volume of return air pulled out. The best results are obtained by pulling that cold air off the floor for a given volume of air. But some are now reversing the natural forces and drawing the return air from points higher up in the room. If their volume is large enough and their downward jet of

sufficient force and volume, they can still do the job. But they are working against the forces of nature when they try it.

H. ZIEL, Detroit, Mich.: We are not always able as engineers to get the space that is required and many times it is an impossibility to get the return air back to the unit from the floor as it has been suggested by Mr. Campbell, and the problem like the jets is the only solution.

I was interested in this paper from a somewhat different angle and that is, for summer operations. What is the effect or what would be the more desirable nozzle to put up in a truss space air outlet when you try to supply outdoor air down to the floor when the air in the truss space of any building is at a relatively high temperature? We have some plants, not very far from the Cincinnati area, where the manufacturing processes produce a temperature condition of 110 F to 150 F in the trusses when the outdoor temperature is 95 F and you try to supply air from outlets in the truss spaces down to the floor. As soon as you introduce the 95 F air at truss space level you tend to aspirate the warm air in the trusses down to the floor line and in place of getting 95 to 97 F temperature at the floor line you will get something like a 103 F, which is most objectionable. I would like to ask the authors of this paper whether they have improvised or could make some recommendations as to the more desirable type of jet to use in supplying warm air down to the floor without aspirating warmer air along with it.

I. A. NAMAN, Houston, Tex.: I would like to ask the authors what the effect would be for multiple outlets instead of only one as was tested. Wouldn't they expect that the intermingling of air from one jet to the other would vary the patterns which they have shown?

AUTHORS' CLOSURE (Dr. Yen): The questions raised certainly show the need for more research. They also give guidance for future research.

Mr. Connor's statement that there is need of research on jets having outlet temperatures 250 F or higher is certainly worthy of consideration. We have done research on such jets from ASME nozzles but not on corresponding jets from the unit heater.

In regard to Mr. Yousifian's question as to which type of outlet would give the optimum temperature distribution in the zone of occupancy, I would point out that the shallow diffuser produced a wider radial spread of temperature effects but a shorter throw than did the annular outlet. What type of outlet should be used will depend upon design conditions, particularly the space to be heated.

Dr. Nottage's question regarding what happens around the small volumes of air throughout the influenced region is a significantly searching question, a satisfactory answer to which would go far toward solving our problem. We have dealt with the problem macroscopically because of the difficulties inherent in the microscopic approach. At any point, there will exist shearing forces due to velocity gradients, and heat transfer due to temperature gradients. It follows that for a complete solution, the equation of motion, the continuity equation, and the energy transport equation must be satisfied. The mathematical and experimental difficulties in the way of such a solution are not small, but as Dr. Nottage pointed out, we are beginning to obtain data that should throw light on the problem and we are engaged in analyzing these data with a view to clarifying our understanding of fundamental relationships. In addition, we are investigating various idealized models of a jet which more nearly represent actual conditions than do the models so far employed. It is hoped that these models will throw light on the influence of outlet factors and boundary conditions with which designers are concerned. We ourselves, are not overly impressed with the order of agreement between test results and the semi-empirical relationships that have been derived from the idealized models of a jet so far employed. In the first place, these relationships have been used to correlate data on only limited aspects of the jet; for example, conditions along the axis of the jet and the down-throw, and the empirical factors have been selected for this purpose. Secondly, from a purely empirical point of view, acceptable correlations may be obtainable using sets of empirical factors significantly different from those

employed. Thus, fairly good correlations of axial velocities are obtainable by using values of n in Equation 1 between ⅓ and unity.

Mr. Phillips would like to know what range of throw factors might be expected and how we would estimate a throw factor for an outlet that has not been tested. So many uncertain factors are involved, for example, the extent of dead area, the degree of whirl, and the velocity distribution at the outlet, that we believe we cannot tell exactly what should be the throw factor for an outlet that we have not tested. A fair estimate probably could be made by comparing the outlet with the outlets that we have tested. Values of throw factor obtained by us range from 0.80 to 1.24. Mr. Phillips commented on the large temperature elevation in the upper part of the room. The principal reason for this would seem to be that the warm air, instead of staying near the floor, ascends to the upper part of the room.

We agree with Mr. Campbell that it is important to find out what happens after heated air leaves the jet proper. We are planning to do this. We have a big room and our jet is relatively small compared to the size of the room. For this reason, the jets we have studied have been relatively free from the influence of wall boundaries. In an actual installation, the walls may have an important influence on the pattern of flow in the room and the distribution of temperature.

We cannot offer a satisfactory solution to Mr. Ziel's problem. However, he might consider the use of outlets with as large a diameter as practicable so that the emitted air will have a velocity as low as feasible. This will not solve the problem, but it should help to reduce the amount of hot air entrained near the outlet.

In respect to Mr. Naman's question, we would expect the pattern of flow of multiple jets to be influenced by the spacing of the jets. If the jets are close enough together to affect each other, then the pattern should be different from what it would be for a single jet.

We appreciate the remarks made by Professor Koestel and shall follow his suggestions when we have the data to do so.

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No. 1563

RESISTANCE OF RECTANGULAR DIVIDED-FLOW FITTINGS

By L. G. MILLER*, C. H. PESTERFIELD**, EAST LANSING, AND R. J. WAALKES†, HOLLAND, MICH.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with the Department of Mechanical Engineering, Michigan State University, East Lansing, Mich.

THE TAKE-OFF is one of the most common fittings of an air-duct system, being second in importance only to the elbow. In spite of this, there have been relatively few systematic investigations of the pressure loss in take-offs‡, and leading reference books contain little information on the subject. The purpose of this research project was to fill one of the gaps in our knowledge, and to determine the pressure loss in typical rectangular fittings designed to accomplish divided air streams.

Fittings of this type are used to divide the air stream in a main into 2 branches, one continuing straight and the other at a right angle to the main, with both branches varying in cross section. To properly connect the 2 branches, engineers have variously shaped the take-off branch, guided undoubtedly by the desire to accomplish the division of the air stream with as little resistance as possible. Some designs use a complete elbow in fitting the branch to the main, others attempt to split the main in a proper place, and still others aim at velocity control by influencing the expansion or contraction of the air stream.

In spite of the multiplicity of possible designs, an analysis indicates that take-off fittings may be divided into 3 general classifications, which are designated in this paper as (1) increaser section with 90-deg elbow, (2) straight section with special elbow, and (3) decreaser section with 45-deg elbow, with each class sub-divided into 3 groups with (*a*) large straight duct and small take-off, (*b*) straight and take-off ducts of equal size, and (*c*) small straight duct and large take-off duct.

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‡ Three ASHAE papers by other investigators dealing with pressure losses in take-offs have been published since the start of this project (References 1, 2, 3).

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Consequently, this investigation has dealt with 9 different take-off fittings, each representing one of the classes and sub-divisions. The size of the main section of each of the fittings of the 3 classes was 20 × 8 in., with 3 fittings having 18 × 8 in. straight duct and 3 × 8 in. take-off duct representing sub-division (*a*), 3 fittings having 10½ × 8 in. straight duct and 10½ × 8 in. take-off duct representing sub-division (*b*), and 3 fittings having 3 × 8 in. straight duct and 18 × 8 in. take-off duct representing sub-division (*c*). As Fig. 1 shows, Class 1 of the fittings (increaser type) expands the approaching air stream to include the sum of the branch widths and employs a full 90-deg elbow with a reasonable throat radius for the

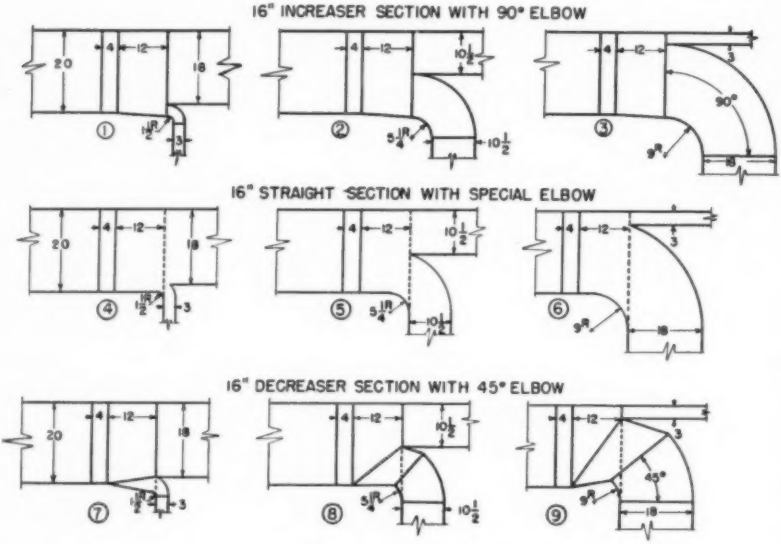

FIG. 1—DETAILS OF FITTINGS TESTED

take-off duct, Class 2 (straight type) uses the principle of dividing the stream at the appropriate point by means of special shape of splitter, and Class 3 (decreaser type) substitutes a 45-deg angle elbow for the customary 90-deg elbow in an attempt to transform the approach velocity in the main to that of the 2 branches.

From a study of the problem it appeared that the pressure loss in the take-off fitting would be dependent upon several factors; namely, the design of the fittings, the respective sizes of the *take-off* and *reduced straight* branches, the total quantity of air being handled through the fittings and the quantity of air being handled by each branch. These 4 factors, therefore, become the variables in the test program.

After a reasonable number of test runs had been made, it was found that the performance of Class 3 (decreaser type) fittings was too erratic to be properly evaluated. Therefore, the present report covers only data for fittings of Class 1 (increaser section 1, 2 and 3) and Class 2 (straight section 4, 5 and 6).

TEST EQUIPMENT AND CALIBRATION

The test equipment consisted of a centrifugal blower connected to a system of ductwork, with quadrant damper, bell-shaped measuring orifice, static taps, and orifices located as shown in Fig. 2.

To keep the duct system from becoming cumbersome and at the same time provide a means for measuring the flow, the resistances in each branch were varied by providing a series of rectangular sharp-edged orifices for each branch. The ducts were provided with U-shaped channels on 3 sides and a slot on the fourth side to permit the orifice plates to be changed externally without disturbing the ductwork.

Control of the total quantity of air being handled by the system was obtained by means of the damper located between the blower discharge and the main measuring orifice. The damper varied this quantity from about 1750 cfm in the full open position to about 200 cfm in the closed position. For flow quantities of less than 200 cfm it was necessary to restrict the blower suction opening.

The velocity pressure at the main measuring orifice was measured by means of 8½ in. (for low readings) inclined draft gages in conjunction with an impact tube and static tap. The static pressures at the tap in the duct just before the take-off fitting and the taps in the 2 branches beyond the fitting were read on similar three, 1½ in. gages. All gages were calibrated in inches of water.

All of the joints beyond the main measuring station (including longitudinal Pittsburgh lock seams) were soldered. The fittings were provided with gasketed flange joints to eliminate leakage and permit interchanging, and the previously mentioned U-shape channels were tightly soldered to the duct to prevent leakage around the edges of the orifice plates. Thereafter, every time a new fitting was put in place, a check was made to determine the extent of the leakage by closing both branches with blanks in place of the sharp-edged orifices. If any appreciable leakage was detected, it was corrected before proceeding with the test.

The sharp-edged orifices in the branches were calibrated by blanking off one of them and causing all of the air to go through the other. The quantity flowing was varied by means of the damper, and the flow rate determined by calculation from the velocity pressure reading at the main measuring orifice. This value was then plotted against the static pressure reading at the tap near the sharp-edged orifice to establish the calibration curve for the orifice.

TEST PROCEDURE

In performing the test work, the outline of the variables previously indicated was followed. A particular style fitting was selected with one of the 3 combinations of branch duct sizes. A series of runs was then made on the fitting, the variable between runs being the combination of orifice sizes in the branches. The variable on each individual run then was the total volume of air flowing which ranged from maximum flow (depending on combination of orifice sizes with damper wide open) down to about 200 cfm. Since there were 4 different orifices for each of the 10½ × 8 in. ducts, the possible combinations made a total of 16 runs for the 10½ × 8 in. by 10½ × 8 in. fittings.

For each run, measurements were made of the velocity pressure at the main measuring orifice; the static pressure in the main duct upstream from the fitting; the static pressure in the reduced straight duct beyond the fitting; and the static pressure in the take-off duct beyond the fitting.

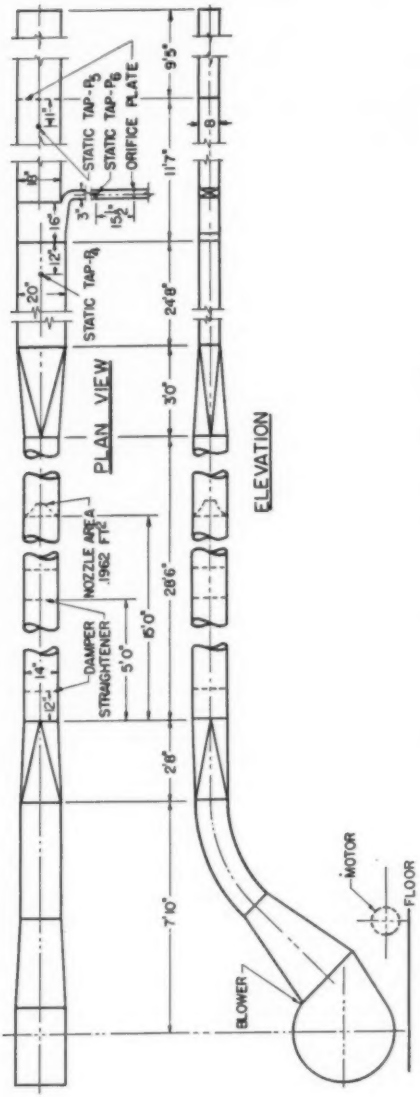

FIG. 2—DETAILS OF TESTING EQUIPMENT

In the actual taking of the data, the velocity pressures greater than 0.5 were read on the 8 in. inclined draft gage and those less than 0.5 in. were read on the ½ in. inclined draft gage after a steady state was reached following the damper adjustment. The static pressures P_4, P_5, and P_6 were read on a 3, 1 or ½ in. gage as required, with the 3 static taps being connected to a common point through a ground glass stopcock for each line. This arrangement permitted the reading of all 3 pressures on the same instrument.

COMPILATION OF DATA

From the test data, calculations were made of the total pressure change from a point in the main duct upstream of the fitting to a point in the branch ducts downstream of the fitting. The friction loss due to the actual length between measuring stations (as obtained from the ASHAE Friction Chart) was then subtracted to determine the loss due to the fitting alone. The results were expressed dimensionlessly as decimal fractions of the velocity pressure corresponding to the mean velocities of flow in both branch and main ducts.

The detailed procedure was as follows:

1. From observed velocity pressure and area of main measuring orifice calculate total quantity of air flowing Q_4.
2. From total quantity of air flowing and area of main duct at static tap P_4 calculate velocity pressure at plane of tap.
3. By addition of calculated velocity pressure and observed static pressure obtain total pressure at static tap P_4.
4. From observed static pressure P_5 and calibration curve determine cubic feet per minute through reduced straight duct Q_5.
5. From Q_5 and area of duct at tap P_5 calculate the velocity pressure at plane of tap.
6. By addition of calculated velocity pressure and observed static pressure P_5 obtain total pressure at plane of tap P_5.
7, 8, 9. By steps similar to 4, 5 and 6 obtain total pressure at plane of tap P_6.
10. By subtraction of the friction loss between P_4 and the fitting determine the total pressure at the upstream side of the fitting. By addition to P_5 and P_6 of the equivalent resistances of the duct on the downstream part of the fitting, determine the total pressure at the downstream side of the fitting.
11. By subtraction of the total pressures at the upstream and downstream points of the fitting, obtain pressure loss due to the fitting.
12. Divide this loss by the velocity pressure at a characteristic cross section, thereby expressing it as a decimal fraction of the velocity pressure. These values are defined in the accompanying tabulation of values of loss coefficients (Table 1).
13. Previous investigations[1] have shown that pressure losses of divided-flow fittings can be correlated with the ratio of branch and main velocities. Hence, the final step was to plot λ_B and $_T\lambda_B$ against V_6/V_4 (See Fig. 5) and to plot λ_S and $_T\lambda_S$ against V_5/V_4. Curves were then faired through the points and subjected to analysis.

Actually, a modification of steps 5 and 8 was necessary since the values of Q_5 and Q_6 as obtained in steps 4 and 7 did not sum to precisely Q_4 (step 1). It was felt that this was due to inability to read the gages accurately to 3 places, and that since the orifice were calibrated against the main orifices in the first place, it would be most consistent to use Q_4 value as being correct. Therefore, the values of Q_5 and Q_6 from the calibration curves were added, and Q_6 divided by the total to obtain the percent diverted in the take-off. This percentage of the correct total cubic

[1] Exponent numerals refer to References.

feet per minute, Q_4, then was a corrected quantity of flow through the take-off Q_{6c}, and by subtraction, $Q_4 - Q_{6c}$, a corrected value of the quantity flowing in the reduced straight duct was obtained Q_{5c}. These two values then were used in steps 5 and 8, and the calculations carried to completion.

The initial plots showed considerable scatter. Detailed analysis indicated the scatter could be reduced considerably by 1) discarding all readings less than 0.010 in. of water; and 2) determining a single, most representative value of pressure loss from the eleven variable cfm runs made with each configuration of the apparatus. The latter was accomplished by fairing a curve through plots of fitting loss *vs* air-flow rate.

RESULTS AND DISCUSSION

In discarding test data having such small pressure differences that the results would be sensitive to slight inaccuracies in measurement, practically all of the data

TABLE 1—VALUES OF LOSS COEFFICIENTS

$\lambda_B =$	Total Pressure Loss (90 deg branch to main) / Velocity Pressure in 90° branch
${}_T\lambda_B =$	Total Pressure Loss (90 deg branch to main) / Velocity Pressure in Upstream Main
$\lambda_S =$	Total Pressure Loss (straight branch to main) / Velocity Pressure in Straight Branch
${}_T\lambda_S =$	Total Pressure Loss (straight branch to main) / Velocity Pressure in Upstream Main

for the straight branches of Fittings 1, 2, 4 and 5 were eliminated. In accordance with conventional laboratory practice[3], the pressure readings were taken downstream of each fitting and corrected back to the junction so as to express the test results on a *no length* basis. Hence, the pressure loss was that caused by disturbances as the flow divided at the junction. Referring to Fittings 1 and 4 in Fig. 1, the 90-deg branch is so small that even though the velocities in the 90-deg branch were varied over a wide range, the maximum flow rate diverted into it was small compared with the total flow rate in the upstream main. The cfm variation in the 18-in. wide straight branch, and hence the variation in velocity, was therefore small. Thus, very little acceleration or retardation of the straight branch flow occurred and, as a consequence, little loss due to flow separation and attendant eddies. The practical significance is that the junction losses of the 4 straight branches were negligible.

Similar reasoning applies to the straight branches of Fittings 2 and 5; *i.e.*, their pressure losses were also essentially zero.

Discarding of data for these 4 branches does not detract from the value of the tests. From a design standpoint, the tests show that the additional pressure loss due to flow disturbances in the straight-through sections of these fittings can be ignored in the design process.

Results for the straight branches of Fittings 3 and 6 are presented in Figs. 3 and 4, respectively. Two curves are shown in each figure; the λ_S curve represents the pressure loss as referred to the straight branch velocity pressure and the $_T\lambda_S$ curve represents the same loss with the velocity pressure in the upstream main as reference. Since the branch and upstream main velocities are the same at $V_5/V_4 = 1.0$, the curves necessarily have the same value at this condition. Comparing the

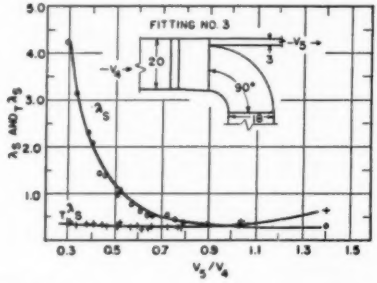

FIG. 3—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR STRAIGHT BRANCH OF FITTING 3

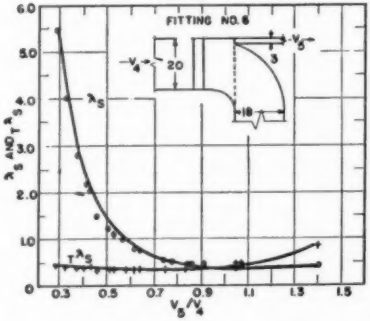

FIG. 4—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR STRAIGHT BRANCH OF FITTING 6

2 figures it is seen that corresponding curves are nearly identical. This is to be anticipated since Fig. 1 shows Fittings 3 and 6 were nearly identical in construction. The principal difference in geometry is that the center of the 90-deg branch radius of Fitting 6 is upstream of the straight branch junction. Apparently any difference in flow disturbances at the inner wall of the 90-deg branches were not propagated across the 18-in. width of duct, or if they were the effects were too small to be of practical significance.

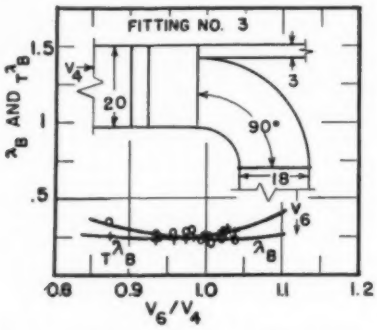

FIG. 5—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR 90-DEG BRANCH OF FITTING 3

Results for the 90-deg branches of Fittings 3 and 6 are presented in Figs. 5 and 6, respectively. The range of velocity ratio variation is narrow. The maximum is limited to a value of 1.1 as a result of the large area of the branch with respect to the main. The range below $V_6/V_4 = 0.85$ was eliminated because of extremely low pressure differences. Over the range of the data, the corresponding Figs. 5 and 6 curves are nearly identical because of nearly similar geometries. Since at a velocity ratio of unity the branch flow is neither accelerated nor retarded, a minimum might be expected there. This is indicated by the results in Fig. 5. Also, acceleration of a flow is an inherently efficient process, so that for some geometries

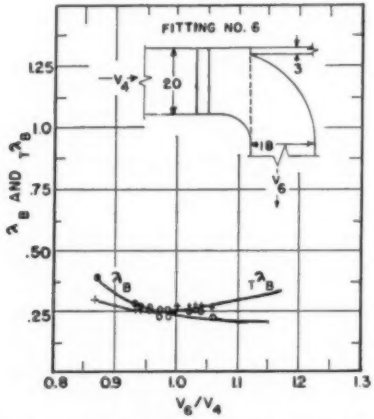

FIG. 6—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR 90-DEG BRANCH OF FITTING 6

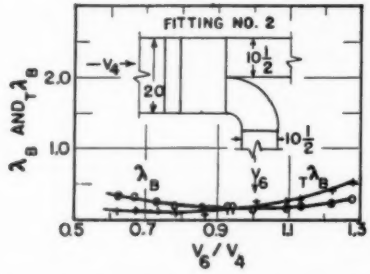

FIG. 7—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR 90-DEG BRANCH OF FITTING 2

the minimum in the λ_B curves might occur at velocity ratios above unity. This possibility is indicated by the λ_B curve of Fig. 6. Since deceleration of the flow is an inherently inefficient process, the occurrence of a minimum value at a velocity ratio less than unity should NOT be anticipated.

At $V_6/V_4 = 1.0$, the 90-deg branch performance should be equivalent to that of an elbow, with perhaps some slight additional pressure loss due to disturbances caused by the straight branch flow. For $V_6/V_4 = 1$, a representative value for both fittings is $\lambda_B = 0.25$. Elbow loss data from the literature[4] yields $\lambda_B = 0.28$. This good agreement validates to some extent the accuracy of the experimental data.

Results for the 90-deg branches of Fittings 2 and 5 are presented in Figs. 7 and 8, respectively. The reduction in branch width from 18 to 10.5 in. yields a wider variation in velocity ratio. The λ_B curve of Fitting 2 has a minimum close to

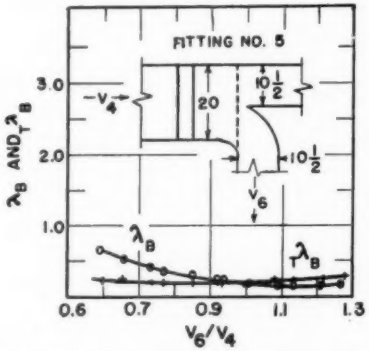

FIG. 8—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR 90-DEG BRANCH OF FITTING 5

$V_6/V_4 = 1.0$ and the Fitting 5 curve at $V_6/V_4 > 1.0$. As discussed, minimums at these velocity ratios are rational. The 16-in. increaser section of Fitting 2 evidently aids the deceleration of the flow because the performance of this fitting is better (lower λ_B values) at $V_6/V_4 < 1$. At $V_6/V_4 > 1$ the performance of Fitting 5 is better, indicating the positioning of the fitting radius further upstream is helpful to an accelerating flow.

This same relative performance should also be expected for Fittings 1 and 4, and comparison of the λ_B curves of Figs. 9 and 10 shows it is indeed the case. Comparison can be made directly in Fig. 11, in which it should also be noted that minimums occur at velocity ratios equal or above unity. Curve C has been obtained from a recent paper[3] and applies to cylindrical 90-deg branch takeoffs connected to main ducts of constant cross sectional area. Since Curve C is restricted to small branch-to-main area ratios, it can only be compared with the Fitting 1

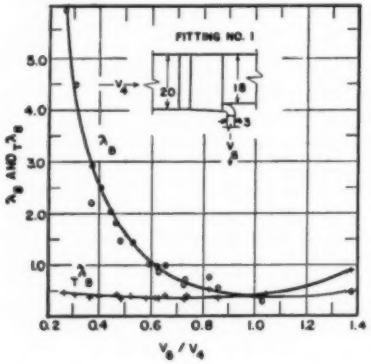

FIG. 9—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR 90-DEG BRANCH OF FITTING 1

and 4 results. Note the similarity in the shapes of the curves and that the λ_B values for the fittings are one-third to one-half of the Curve C values at the same velocity ratio. Thus, Fig. 11 shows Fittings 1 and 4 are considerably more effective in diverting the flow into the 90-deg branch. However, their effectiveness is obtained at the sacrifice of fabrication and installation costs. Moreover, at velocity ratios considerably above unity, Curves A and B will eventually rise and cross Curve C. The conclusion is therefore that the geometries of Fittings 1 and 4 are more favorable (with respect to pressure loss only) for low velocity ratios, and the geometry of the cylindrical type of divided-flow Fitting 3 is more favorable for high velocity ratio applications.

The fittings of Fig. 1 are ordinarily used in so-called *Trunk Duct Systems*[5]. In these systems the 90-deg branch velocity is usually below the upstream main velocity. The practical range of the data presented is therefore in the region of velocity ratios less than 1.0. Inspection of the $_T\lambda_B$ and $_T\lambda_S$ curves of Figs. 3 to 10 discloses they are nearly horizontal in this region. As a consequence, for all

practical purposes, the pressure loss can be expressed as a certain fraction of the velocity pressure in the *upstream main*. If this is done, the results are also easily expressed in *equivalent diameter of upstream main* if a value of friction factor f is assumed. The pressure loss of the straight branch is, in inches of water,

$$H = {}_T\lambda_B \left(\frac{V_T}{4005}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

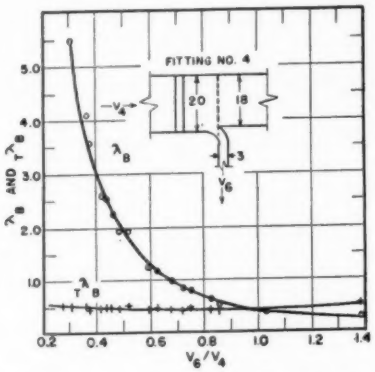

FIG. 10—PRESSURE LOSS IN TERMS OF VELOCITY PRESSURES FOR 90-DEG BRANCH OF FITTING 4

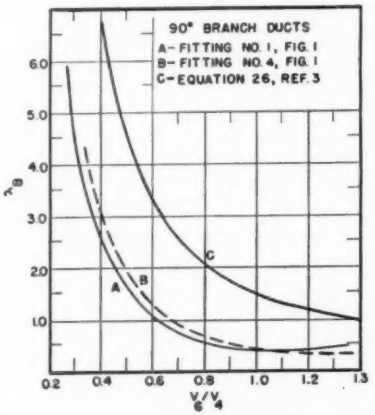

FIG. 11—COMPARISON OF TEST RESULTS WITH THOSE AVAILABLE FOR CYLINDRICAL, BUTT-TYPE TAKEOFF

where V_T is the mean velocity in the upstream main. Since the pressure loss in a straight pipe is

$$H = f\frac{L}{D}\left(\frac{V_T}{4005}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Equating these, there is obtained:

$$L/D = {}_T\lambda_S/f \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

For the 90-deg branches, ${}_T\lambda_S$ in Equation 3 is replaced by ${}_T\lambda_B$. Based on these equations, design data for the usual range of velocity ratios are presented in Table 2.

TABLE 2—DESIGN DATA FOR FITTINGS WHEN BRANCH-TO-MAIN VELOCITY RATIO IS LESS THAN 1.0

FITTING NO.	BRANCH	LOSS IN UPSTREAM MAIN VELOCITY HEADS	L/D, EQUIVALENT DIAMETER OF UPSTREAM MAIN DUCT[a]
1	Straight	0	0
1	90°	0.40	20
2	Straight	0	0
2	90°	0.16	8
3	Straight	0.30	15
3	90°	0.26	13
4	Straight	0	0
4	90°	0.44	22
5	Straight	0	0
5	90°	0.20	10
6	Straight	0.36	18
6	90°	0.26	13

[a] Based on a friction factor f of 0.02.

If the designer prefers to use the loss in terms of the *branch* velocity pressure, the appropriate λ_B or λ_S curve can be used directly. Alternatively, ${}_T\lambda_S$ in equation 3 can be replaced by λ_B or λ_S to obtain the pressure loss in *equivalent diameters of branch duct.* It should be noted, however, that the equivalent diameter of *branch duct* is not a constant but varies markedly with velocity ratio.

CONCLUSIONS

1. Performance of the fitting termed the *decreaser section with 45-deg angle* was erratic; test data for this type were therefore discarded.
2. Pressure losses at the junction of the straight branches were negligible for branch-to-main area ratios above 0.5.
3. The *increaser section with 90-deg elbow* was more effective than the *straight section with special elbow* at velocity ratios less than about 1.0, and less effective at higher velocity ratios.
4. At velocity ratios of unity, the test results were in reasonable agreement with published elbow loss data.

5. The 90-deg branch losses were only one-third to one-half of those reported for cylindrical butt-type takeoffs, which is to be expected for the range of velocity ratios in which the tests were conducted.

6. Over the practical range of velocity ratio for the type of fitting tested, the pressure loss can be expressed as a certain fraction of the velocity pressure in the *upstream main* or, to a good approximation, by a single value of equivalent diameters of *upstream main duct.*

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DISCUSSION

W. O. HUEBNER, New York, N. Y.: As chairman of the TAC on Air Distribution, I am extremely interested in the research papers which have just been presented. Therefore, I ask your permission to say a few words about Professor Helander's and Dr. Yen's paper on *Jets* and Professor Miller's, Professor Pesterfield's and Mr. Waalkes' paper on the *Resistance of Rectangular Divided-Flow Fittings.*

It is a trite but just observation that room air distribution problems are very complicated, and that we also do not know enough about the behavior of air confined in ducts. Therefore, we are very happy to have learned a little more about it.

There are two brief points which I would like to stress. First, I would like to draw your attention to the fact that the formulas in the first paper are illustrated by an example. This should make the paper of particular value to everyone concerned with the problems discussed in the paper.

Secondly, and Mr. Waalkes has already pointed this out, it is gratifying that the paper on Divided-Flow Fittings confirms the results obtained by other investigators, for example, those obtained by Dr. Gilman at the University of Illinois.

In connection with a question which was asked by a gentleman from Houston, I am happy to state that Professor Koestel's paper on *Air Velocities in Two Parallel Ventilating Jets* is ready for publication and is expected to be presented at the coming Semi-Annual Meeting of the Society.

W. M. ROWE, Cincinnati, Ohio: Table 1 entitled *Values of Loss Coefficients* indicates *total* pressure loss in the test section. Fig. 2. *Details of Testing Equipment* indicates that the pressure drops across the test sections were taken with static pressure taps at P4, P5 and P6. Should not the coefficients listed in Table 1 then be indicated in terms of static pressure rather than total pressure?

AUTHORS' CLOSURE (Mr. Waalkes): I would like to thank Mr. Rowe and Mr. Huebner for their comments. Perhaps I should also thank Mr. Huebner for making comments which didn't require an answer.

To answer Mr. Rowe's question, which is well-taken, let me explain that time didn't permit going through the complete calculation procedure. Actually in describing the test setup I mentioned static taps at 3 points. The complete procedure involved the

reading of the static pressure at these points; from known flow quantities (as determined from the main measuring nozzle for the total flow and from the calibrated orifices for the 2 branches) we then calculated what would be the average velocity at these 3 points, and from these velocities the average velocity heads. The sum of the observed static pressures and calculated velocity pressures then gives total pressures. So we were working with total pressures throughout, and the loss coefficients given in Table 1 are correctly expressed there with total pressure losses expressed as fractions of velocity head. This was not made clear in the abbreviated presentation of the paper.

I would just like to take one more minute to emphasize that the expressing of the resistance of the branches of these fittings as a fraction of the velocity head of the upstream main appears to be preferable to using the branch velocity head. In Dr. Gilman's paper it was actually done both ways, and, as I recall the paper, it gave no specific recommendation as to which was the better reference. I believe he did point out in that paper, as would be evident from our curves also, that the use of the branch velocity head, as a reference, could be misleading. You will remember that the λ_B (lambda B) and λ_S (lambda S) curves for the small branches went almost to infinity. At the corresponding velocity ratios it's not so readily apparent that as the velocity ratio decreases the branch velocity itself also decreases. Therefore, even though the loss expressed as a fraction of the branch velocity head is high, the actual total pressure loss in inches of water may, in some instances, be less than at the higher velocity ratios. So it is misleading. It looks like a bad condition whereas our curves were plotted for the same total loss in inches of water in either case. It's just a matter of how it is expressed, and expressing it as a fraction of the approach velocity head not only gives a lower value but also a more nearly constant value, and therefore, as we see it, is the better way of doing it.

There is one other aspect which makes the selection of the upstream main velocity head the better of the two, and this is that in the initial design stages one normally will know the air quantities, the cfm being handled through various sections of the main trunk. One will also know the cfm wanted through the branch, but the velocity of the branch may vary since it is necessary to take into consideration the size of the outlet, the desirable outlet velocity, throw from the outlets, and other factors, which may affect the sizing of the branch duct. For instance, with an outlet of high resistance one may try to compensate by reducing the duct loss by using a larger size duct. Therefore, though the cfm may stay the same, the velocity may vary depending on the final design.

If one has used data in the design process which is dependent on the velocity of the branch, these data will be changed when that velocity is changed; whereas if the cfm remains unchanged, the cfm in the main and consequently the velocity of the main is unchanged and if that has been used as a reference the design data are still correct.

Again I would like to say that comments are invited. This paper did not appear until the January issue of the Journal Section so perhaps many haven't had a chance to look it over. After having done so, we would like to have comments, and we hope the information will prove useful.

1564

No. 1564

EVALUATION OF EQUIPMENT NOISE

By H. C. HARDY* AND D. E. BISHOP**, CHICAGO, ILL.

THE SOUND level meter, invented approximately 25 years ago, is an instrument which measures the minute alterations of pressure in the atmosphere that we call sound waves. The magnitude of such pressure fluctuations is small. A pressure fluctuation of about one millionth of one atmosphere (one dyne per square centimeter or approximately 0.0004 in. of water) gives a sound pressure level of 74 db. Fig. 1 gives an idea of the magnitude of these sound waves.

The meter is the basic instrument for measuring sound, but a reading of the meter alone is inadequate to determine the effect of sound just as a measurement of temperature at a point on a wall presents an inadequate measure of the heat input into the room and the effect it has on the people in it. What is really needed in the acoustical case is knowledge of:

(*a*) How much sound energy is emitted by the sound source and where the sound energy is going; and
(*b*) How an individual reacts to this sound energy, in particular how loud it is to him.

In a manner similar to that used to compute the aerodynamic energy flow and pressure drops in a hydraulic system, it is possible to follow the acoustic energy transfer by obtaining the acoustic power output of the source and the losses along various paths. Power output and losses, however, will vary considerably with frequency, and the human reaction can only be interpreted in terms of the acoustic energy distribution with frequency. It is essential, therefore, that measurements be distributed over several frequency bands.

It should be evident that, only when the knowledge requirements (*a*) and (*b*) given in the preceding paragraph are established, is it possible to design the systems intelligently or efficiently. A considerable part of this paper explains how these data can be obtained for air flow systems.

RELATIONS BETWEEN SOUND PRESSURE AND ENERGY

A more basic unit than sound pressure, but one which is not directly measured is sound intensity (I), the amount of energy per second passing through a unit

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area. For a plane wave or a spherically diverging sound source in open air (free field) the intensity (I_F) is related to the sound pressure (p_F) by the relation

$$I_F = p^2_F/\rho c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

p = density of the air.

c = the velocity of sound at room temperature.

ρc = 42 cgs units.

When the sound is completely random in a room (a condition which is approximately true in any large room at high frequencies, especially when the measure-

	DECIBELS	THRESHOLD OF FEELING	INTENSITY UNITS	SOUND PRESSURE DYNES/SQ.CM.
VERY LOUD DEAFENING	120	THUNDER, ARTILLERY	1,000,000,000,000	200.
	110	NEARBY RIVETER ELEVATED TRAIN	100,000,000,000	
	100	BOILER FACTORY	10,000,000,000	20.
	90	LOUD STREET NOISE NOISY FACTORY	1,000,000,000	
	80	TRUCK UNMUFFLED POLICE WHISTLE	100,000,000	2.
LOUD	70	NOISY OFFICE AVERAGE STREET NOISE	10,000,000	
	60	AVERAGE RADIO AVERAGE FACTORY	1,000,000	0.2
MODERATE	50	NOISY HOME AVERAGE OFFICE	100,000	
	40	AVERAGE CONVERSATION QUIET RADIO	10,000	0.02
FAINT	30	QUIET HOME OR PRIVATE OFFICE	1,000	
	20	AVERAGE AUDITORIUM QUIET CONVERSATION	100	0.002
VERY FAINT	10	RUSTLE OF LEAVES WHISPER	10	
	0	SOUND PROOF ROOM THRESHOLD OF AUDIBILITY	1	0.0002

FIG. 1—TYPICAL OVERALL SOUND PRESSURE LEVELS FOR COMMON SOUNDS

ment is averaged over a band of frequencies) the relationship is

$$I_R = p^2_R/4\rho c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In any particular space in which the geometry is not simple, the relationship between the sound pressure and the intensity cannot usually be calculated[1].

The total power (P) (energy per second) emitted by a spherical sound source radiating in free space would be

$$P = 4\pi r^2 I_F = 4\pi r^2 (p^2_F/\rho c) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

[1] Exponent numerals refer to References.

where

r = the distance from the source to the point where the sound pressure p_F is measured.

When the sound source is not symmetrical, as is often the case, the power can only be found by numerical integration of measurements made in many directions.

In a random sound field (an enclosed reverberant space) this integration can be obtained automatically at a single measurement position. After the sound is turned on, a condition of equilibrium is soon reached where the power added by the sound source equals the power absorbed in the room. In such a random field, the manner in which the sound field changes with distance is shown in Fig. 2. At first the sound level decreases, as given by Equation 3, but soon the direct sound

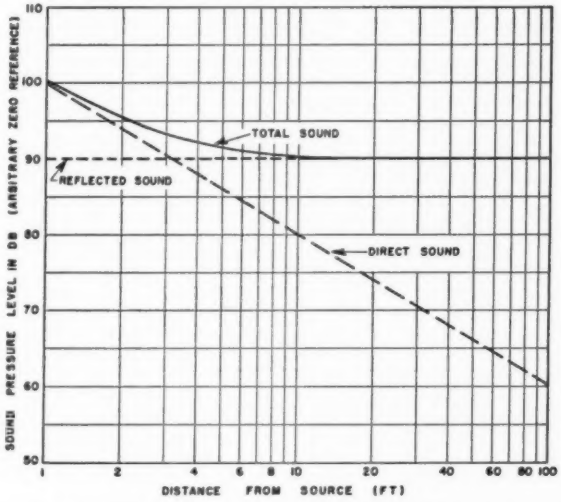

FIG. 2—VARIATION OF NOISE LEVELS WITH DISTANCE FROM A SIMPLE SOURCE IN A REGULARLY SHAPED ROOM

is lost in the energy reflected from other surfaces, and the intensity becomes almost uniform and independent of the distance from the source[2]. The random level in the room will depend only on the total absorption; the higher the absorption, the lower the level.

For the case where there is little sound absorption in the room (a hard-walled room) and at distances of several feet or more from the source:

$$P = I_R A = p^2_R A/4\rho c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

where

p_R = the random sound pressure level.

A = the total sound absorption in the room, equal to:

$$A = A_o + \Sigma S_i \alpha_i. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where

A_o = the air absorption.
S_i = the area of a surface.
α_i = the absorption coefficient of that surface.

RANDOM FIELD MEASUREMENTS

The foregoing statements were necessary to give an understanding of how the sound power emitted by a heating and ventilating unit can be evaluated from sound level measurements. It can be seen from Equation 3 that generally in free space

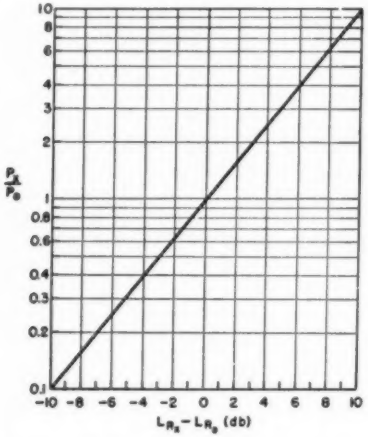

FIG. 3—COMPARISON OF THE SOUND POWER OUTPUT RATIO BETWEEN UNKNOWN AND STANDARD SOURCES WITH THE DIFFERENCE BETWEEN REVERBERANT SOUND PRESSURE LEVELS

the sound level measurements vary considerably with distance and usually with angle. It is suggested, therefore, that the sound power be measured when possible in random field conditions. These are conditions which exist approximately in most indoor spaces.

It is proposed, therefore, that standards on sound output be written in terms of sound power emitted into a random space. It will be necessary in the general case that some constants of the room and measuring equipment be known. However, in the specific case it will not be necessary to know the absorption of the measuring room or the calibration of the sound level meter since every measurement can be a comparison measurement against a standard known source. Thus the P_x of the unknown will be related to the power of the standard P_o, *viz*

$$P_x/P_o = p_{Rx}^2/p_{Ro}^2 . \quad (6)$$

$$10 \log P_x/P_o = 20 \log p_{Rx} - 20 \log p_{Ro} = L_{Rx} - L_{Ro}$$

$$P_x/P_o = \text{antilog } 1/10\ (L_{Rx} - L_{Ro}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where

L_{Rx} and L_{Ro} = the random sound levels in the same room when sound sources P_x and P_o are interchanged.

Fig. 3 is a chart which gives P_x/P_o as a function of $L_{Rx} - L_{Ro}$.

The standard unit can be either a comparison machine which is run under standard conditions or it can be an artificial sound source created by playing a particular kind of noise through a loudspeaker unit. The calibration can be obtained from either elaborate free-field measurements or from measurements in a reverberation

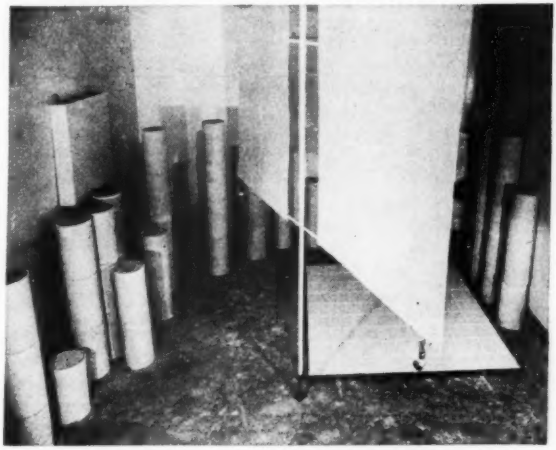

FIG. 4—INTERIOR OF THE RIVERBANK REVERBERATION CHAMBER, A ROOM USED TO DETERMINE THE SOUND ABSORPTION OF ACOUSTICAL MATERIALS

room used for determining the absorption of acoustical materials[3]. Fig. 4 shows one such laboratory.

MEASUREMENT OF FANS

Let us see, for example, how this method could be applied to the measurement of the acoustical power output of industrial fans. Fig. 5 shows a possible test setup for measuring such fans. The fan is placed in a fan room A which is connected to a measurement room B, each of which is made as acoustically alive as possible. It is desired to measure the noise power produced in the intake, the exhaust, and in the fan room itself. The noise radiated by the motor and fan mechanism is measured in room A and compared with a standard source. The exhaust noise is measured by attaching a portable noise attenuating labyrinth to the intake. This is later attached to the exhaust so that the intake noise energy can be measured.

Provision is made for the control and measurement of air flow at either intake or exhaust. In general, only one sound level measurement need be made for each measurement condition, although measurements at several positions may be used at low frequencies to get a better average.

How to Use Noise Power Data

Given the amount of sound power produced by a fan—which, for reasons given later, needs to be measured at a number of frequencies—the useful acoustical calculations can be made. For instance, knowing the acoustic power produced in the exhaust duct, calculations can be made of:

1. The amount of sound power taken off at each branch point. (Generally, the power will divide proportionally to cross-sectional area).
2. The acoustic power absorption of acoustical lining.

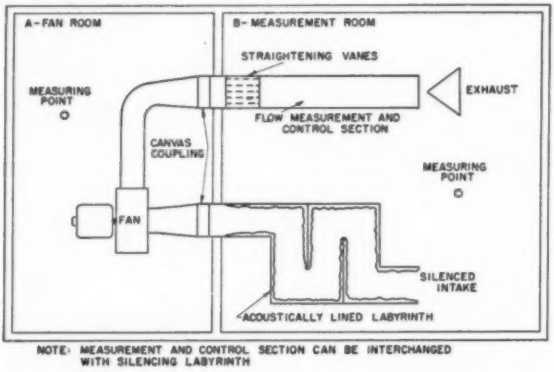

FIG. 5—POSSIBLE TEST ROOM ARRANGEMENT FOR DETERMINING THE SOUND POWER OUTPUT OF INDUSTRIAL FANS

3. The effect of change of cross section such as plena, lined or unlined.
4. The effect of turns.
5. The loss due to radiation through duct walls.

Finally, the amount of acoustical power arriving at each grille or outlet can be obtained. Knowing the acoustical power radiated at the outlet, the sound pressure in the room can be obtained by using either Equations 3 or 4 depending on whether information near the source or at a distance is desired.

Furthermore, if noise is regenerated in the system (for instance at the grille), the noise power so created can be evaluated by separate measurements and added at the proper place to the total power output.

Many of the data needed to give the answers in the foregoing list are still not available, chiefly because up to now there has not been a satisfactory plan on which they can be organized. With the approach outlined previously it is believed that useful working data could be obtained on various practical systems with a relatively simple measurement program.

IMPORTANCE OF FREQUENCY MEASUREMENTS

Up to this point little has been said about making sound pressure level measurements at different frequencies. It has been common practice in the past in the heating and ventilating profession to depend only on overall sound level measurements. These are only a little more than useless for 4 reasons: viz,

1. Pieces of equipment may yield similar overall sound levels, yet their distribution of sound energy with frequency may be greatly different.
2. Our ears hear high frequencies more easily and call them louder.
3. There is a great difference in the ease of propagation of low frequencies through walls and ducts compared to high frequencies.
4. The levels of masking environmental or background noises differ widely at various frequencies.

Fortunately, there is available frequency analyzing equipment which is connected to and used to supplement the sound level meter. By far the most useful analyzer is an octave band filter which is sufficient to analyze nearly all noise measurements that need to be applied to heating and ventilating problems. This filter divides the acoustical spectrum up into slices each of which is an octave wide (an octave is a range of frequencies of two to one). The official standard octaves are: 37.5–75, 75–150, 150–300, 300–600, 600–1200, 1200–2400, 2400–4800, and 4800–9600 cycles per second (cps). The state of the art has grown to such an extent that there is little reason at present for making only overall sound level measurements.

CONVERSION OF OCTAVE BAND LEVEL TO LOUDNESS MEASUREMENT

One of the reasons why octave band level measurements are important is that they can be converted into subjective units of loudness, if it is desired to do so. It should be remarked that the conversion from physical to subjective units is never made unless one can conceive of a person actually listening at that point. For instance, calculation of the loudness inside a ventilating duct would generally be ridiculous.

The unit of loudness is a *sone* and unlike the decibel it is an arithmetic unit such that 10 sones are twice as loud as 5 sones. A graph paper for converting octave band levels to loudness is shown in Fig. 6[4]. The graph gives the loudness of the noise in each octave. The total loudness is the sum of the individual loudnesses. Not only will this scale of loudness rank various units of the same kind, but it can be used with reasonable assurance to compare the loudness of, for instance, a room air-conditioning unit with a refrigerator, or a typewriter with a fan. Final evaluation on a loudness basis also shows what portions of the frequency range are most important to consider if further quieting is needed.

It is possible, therefore, by referring to a standard distance or to standard room conditions to rank various air-conditioning or heating units by a single number system, and it is further possible to write standards using this notation.

EXAMPLES OF APPLICATION

This paper will be concluded with 2 examples to illustrate the use of the method.

In the first example, we would like to determine the sound levels in an office caused by the ventilation supply for the office. The supply is a branch duct,

attached to a main supply duct served by a centrifugal fan delivering approximately 8000 cfm. The first line of Table 1 shows the difference in the reverberant sound pressure levels between the fan exhaust and a standard source, as measured in a setup similar to that of Fig. 5. The known power output of the source is given in line 2; the sound power output, in line 3, can be determined by use of Fig. 3. For the case of an area reduction of 10 to 1 between main and branch ducts, the sound power in the branch duct is that given in line 4 (assuming the sound power reduction is proportional to the area ratio.) Line 5 gives the sound pressure level in the branch duct, 12 in. x 18 in. in size, calculated by use of Equation 1. Line 6 gives the typical sound absorption to be found in an office 20 x 30 x 8 ft. Equation 4 can now be used to calculate the sound pressure levels in

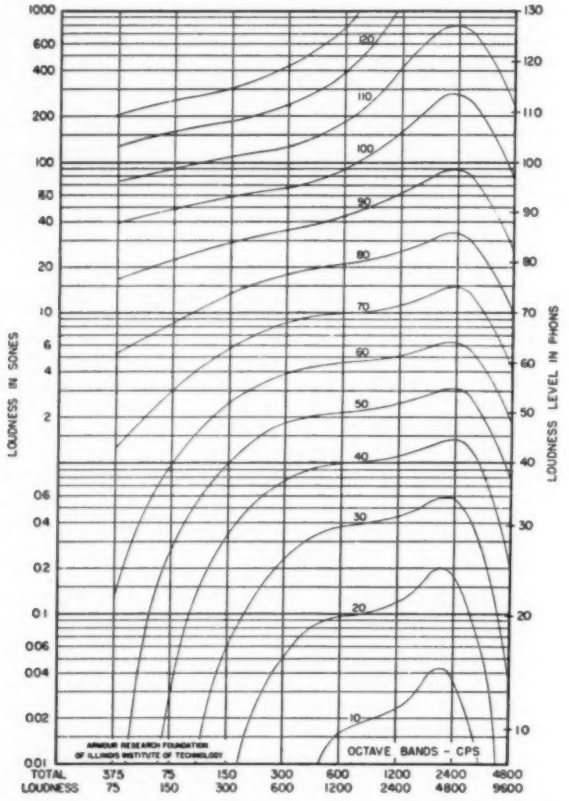

FIG. 6—LOUDNESS GRAPH PAPER, USED TO EVALUATE OCTAVE BAND SOUND PRESSURE LEVEL MEASUREMENTS IN TERMS OF LOUDNESS

TABLE 1—EXAMPLE OF TYPICAL NOISE ANALYSIS OF SIMPLE VENTILATION SYSTEM

	OCTAVE BANDS OF FREQUENCY, CYCLES PER SECOND (CPS)							
	37.5 75	75 150	150 300	300 600	600 1200	1200 2400	2400 4800	4800 9600
1. $L_{RX} - L_{R0}$ (db)	+10	+5.5	+0.5	−1	−1.5	−4	−7	−10
2. P_0 (microwatts)	120	180	200	160	80	35	10	3
3. P_x (microwatts) – Main Duct	1200	640	250	125	40	14	2	0.3
4. P_x (microwatts) – Branch Duct:	120	64	25	12.5	4	1.4	0.2	0.03
5. SPL (db)—Branch Duct	90	87	83	80	75	70	62	54
6. Absorption in office (sabins)	100	140	180	220	240	260	280	300
7. SPL (db) – in office	78	73	68	64	59	54	45	37
8. Loudness (sones)	3.7	4.1	4.7	5.2	4.2	3.3	2.1	0.1

Total loudness of unit ≅ 27 sones

TABLE 2—EXAMPLE OF NOISE ANALYSIS FOR ¾-TON AIR-CONDITIONING UNIT

	OCTAVE BANDS OF FREQUENCY, CYCLES PER SECOND (CPS)							
	37.5 75	75 150	150 300	300 600	600 1200	1200 2400	2400 4800	4800 9600
1. $L_{RX} - L_{R0}$ (db)	−21	−13.2	−12	−14	−22	−24.5	−28	−34
2. P_0 (microwatts)	120	180	200	160	80	35	10	3
3. P_x (microwatts)	0.95	8.0	12.2	6.4	0.50	0.125	0.016	0.0012
4. Absorption in bedroom (sabins)	60	80	100	110	120	130	140	160
5. SPL (db) in bedroom	59	67	68	64	53	47	37	25
6. Loudness (sones)	0.1	2.0	4.5	5.2	2.7	1.9	1.1	< 0.1

Total loudness of unit ≅ 18 sones

7. Typical residential noise levels at night (db)	42	44	42	38	32	26	20	<15
8. Loudness (sones)	<0.01	0.08	0.41	0.60	0.46	0.26	0.18	< 0.01

Total loudness of background ≅ 2 sones

the office; these values are given in line 7 of Table 1. (These sound levels hold for positions more than about 6 ft from the grille opening.) The loudness values can now be determined by use of Fig. 6 and are shown in line 8. The total loudness is 27.4 sones, the summation of the loudness in each octave band.

This example omits such effects as the sound reduction along the unlined duct, or that due to turns or bends, or grille noise. However, these necessary refinements may easily be taken into account once the acoustic performance of these parameters has been established.

Values of typical office noise spectra can now be compared with lines 7 and 8. From this comparison a quantitative estimate of the noise reduction requirements can be determined for the office supply system.

For the second example we will consider the noise output of a ¾-ton air conditioner. The difference in reverberant sound pressure levels between the air conditioner sound output and the standard source are given in line 1 of Table 2. For this comparison, made in a hard-walled room, the noise from the rear of the unit, which would radiate to the outside of a room in normal use was reduced by a silencing labyrinth similar to that indicated in Fig. 5. Line 3 of Table 2 shows the power output of the unit. Typical absorption values for a medium sized bedroom are shown in line 4. The reverberant sound levels in this room, calculated

from Equation 4 are given in line 5, while the loudness values are shown in line 6. Typical background sound pressure levels and loudness values for a residence at night are given in lines 7 and 8. It can be seen that since the unit sounds over 8 times as loud as the background noise, it will very likely be objectionable and probably should be quieted considerably. Examination also shows that noise reduction is most needed in the frequency region from 150 to 1200 cps.

CONCLUSION

This paper has presented the elements of the modern acoustical engineering techniques for handling noise measurement and control in ventilating systems. By converting sound pressure measuring data into acoustic power and its relative contribution in octave bands of frequency, it is possible to compute acoustical energy flow paths and the final subjective loudness heard by the listener.

This system of calculation has already been used by the Armed Forces Laboratories and is the basis of new standards which are being written for measurement of equipment noise. Before it can be of ultimate use to the engineering profession, however, research will have to be undertaken to obtain working data in the form of charts and tables. Such data as the following should be obtained:

1. Typical sound power spectra of fans and other air movement equipment. (Specific spectra would be the responsibility of the manufacturer.)

2. Working data on the effect of turns, branch-points, and plena on noise power propagation.

3. The noise power transmission through typical duct walls.

4. The effect of acoustical duct lining on sound attenuation for various size ducts at various frequencies.

5. The regeneration of noise by turbulence at bends, grilles, and other elements of the system as a function of velocity. (These data will be particularly important in high velocity flow systems.)

6. The acoustic absorption properties of typical rooms or spaces in which ventilation unit outlets are placed.

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2. A Proposed Loudness Efficiency Rating for Loudspeakers, and the Determination of System Power Requirements for Enclosures, by H. F. Hopkins and N. R. Stryker (*Proceedings of Institute of Radio Engineers*, Vol. 36, 1948, pp. 315–35).

3. Direct Measurement of the Efficiency of Loudspeakers by Use of a Reverberation Room, by H. C. Hardy, H. H. Hall, and L. G. Ramer (*Proceedings of National Electronics Conference*, Vol. 8, 1953, pp. 99–107).

4. A Loudness Chart for Octave-Band Data on Complex Sounds, by F. Mintz and F. G. Tyzzer (*Journal Acoustical Society of America*, Vol. 24, 1952, pp. 80–82).

DISCUSSION

C. H. ALLEN* AND L. L. BERANEK*, Cambridge, Mass. (WRITTEN): Three principal topics considered in the authors' paper might be listed as follows:

* Bolt Beranek and Newman, Inc.

1. *Noise Specification*: Noise cannot be specified nor described by a value of the overall sound pressure level. The sound must be broken into at least eight octave bands and sound pressure levels determined for each. With this the discussers agree completely and will present some additional evidence in its support.

2. *Noise Source Measurement*: A sound source can be described by determining for each octave band (a) the total sound energy emitted and (b) the relative amount of sound radiated in any direction, i.e., the directivity pattern of the source. With this we are again in agreement, but the reverberation method of measuring or comparing sound sources does not give information concerning the direction of the radiated sound, and, therefore, this method is limited to situations where directivity is unimportant. We will discuss the extent of these limitations and indicate some of the alternatives which are available.

3. *Noise Criteria*: A noise criterion is determined by the way an individual reacts to the noise being considered in particular to its loudness. This is true, but subjective loudness is not the only determining factor which is of interest. One must consider the type of activity being carried on by the individual and the purpose of the noise control when deciding upon a criterion. We will present some additional criteria of practical interest.

Each of these 3 topics will now be discussed in more detail.

Noise Specification: Elimination of noise in general costs money. It can be carried profitably only to the extent where it gives competitive advantage. Additional expenditure to provide noise reduction beyond what is needed reduces competitive advantage by increasing cost to the consumer. It is necessary to know, therefore, what noise is causing trouble and from this to determine the least expensive treatment which can be applied. In order to specify a noise level or to determine a required noise reduction for a piece of equipment it is necessary to specify the maximum amounts of noise permissible throughout the audible spectrum and, as a practical minimum, it generally has been found necessary to specify the noise levels in each of 8 octave bands from 20 to 10,000 cps.†

By way of illustration it might be pointed out here that, in THE GUIDE, overall values are given for acceptable noise levels in various types of rooms and vehicles. These overall values are not sufficient to allow for suitable noise control. For example, a broadcasting studio is specified as having a sound level of 20 to 30 db on the flat response network. This level would in fact be very difficult and expensive to obtain if indeed it were possible. For a broadcasting studio the sound pressure level in the three octave bands 600 to 1200, 1200 to 2400, 2400 to 4800, should average less than 15 or 20 db, but the sound pressure level in the lower frequency bands may be allowed to increase gradually so that in the lowest band, 20 to 75 cps, the sound pressure level may be as high as 50 db without causing any observable disturbance, as will be explained later when considering criteria. This is 20 to 30 db above that indicated in THE GUIDE. To obtain 20 to 30 db of attenuation in the first band would require a tremendous and needless expense. With the low frequencies present, a sound level meter would read 50 db on the flat scale and the studio would be completely satisfactory. However, a studio which might give a reading of only 30 db but which had most of the noise in the higher frequency bands would be entirely unusable. This is only one example but it illustrates the point in question.

The relative intensities of various frequencies in a noise spectrum are of prime importance. An analysis of noise into at least 8 octave bands is necessary from the standpoint of economics if for no other reason. Specifications for noise levels should be written to include such an analysis.

Noise Source Measurement: A noise source can be described by determining for each octave band both the total energy emitted and the directivity pattern of the source. A determination of the total energy alone is sufficient only when the source radiates uniformly in all directions or when the source is in a reverberant room and all regions of

† It is common practice to extend the first and last octave bands to include somewhat more than one octave, i.e., 20 to 75 cps and 4800 to 10,000 cps.

interest are sufficiently far from the source that the multiplely reflected sound of the reverberant field is of higher level than the direct sound from the source.

In general, these conditions are rarely met in practice. Most sources have linear dimensions comparable to, or larger than a wave length of sound and radiate non-uniformly. A duct opening is one such source. Sound tends to beam in the direction normal to the face of the duct opening. The direct sound level in some frequency bands measured at remote points equidistant from the center of a large duct opening may exceed by as much as 15 db the value which would exist if the source radiated uniformly in all directions.

Most rooms have sufficient acoustical absorption that the reverberant sound level is below the level of the direct sound for several feet from the source especially in the direction of maximum radiation. A knowledge of the radiation pattern will frequently aid in the location of duct openings or in the orientation of unit equipment so as to remove the direction of maximum radiation from the principal occupied area. The simple expedient of relocation of a duct opening from a ceiling to a side wall so that the sound is beamed over the heads of occupants of the room may reduce the amount of sound attenuation needed in a ventilating system by 6 to 10 db in some octave bands. To this same extent a calculation of sound pressure level in a room neglecting effects of directivity of the source may be in error and predict a level much lower than would actually exist in some locations in the room. A detailed discussion of noise radiation from ventilating systems has been published.[1,2]

The radiation pattern and total sound output for various types of ventilating and air-conditioning unit equipment must be measured experimentally since sound is radiated not only from exhaust and intake openings but also from vibrating panels and supporting members. Care must be exercised in such measurements to insure that the equipment is operating in a manner duplicating as nearly as possible the installation which is to be used because the proximity of a wall or other large sound reflecting surface will in general significantly affect the directivity and may either increase or decrease the total sound output by changing the acoustic load on the equipment.[3,4] The measurement of total sound output in a reverberation room does not avoid this difficulty because the mounting condition alters the total sound output of a piece of equipment as well as the directivity pattern.

When only the total sound power is needed the reverberation room technique is satisfactory if proper consideration is given to the mounting of the equipment. However, care must be taken to be sure that the microphone reading is truly an average value. Sound pressure levels measured at several points extending over a distance of at least one wave length should be averaged because individual values may depart from the average by several db, especially at low frequencies due to the presence of standing waves. Reverberation room techniques are being studied by a committee of the American Standards Association with a view toward determining practical limits for probable accuracy of results under specified conditions of frequency range, standing wave ratios, and room dimensions.

As a means of duplicating actual installation mounting conditions and to enable the measurement of directivity, some manufacturers are now employing specially designed rooms with a hard floor and one or two hard walls; the remaining walls and ceiling are covered with anechoic wedges to eliminate reflections. In such a room directivity can be measured by scanning or can be approximated as closely as needed by the use of a suitable number of fixed microphone positions. The total noise output may be calculated by integrating over a sphere, hemisphere, quarter sphere, or octant, whichever

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[1] Apparatus and Procedures for Predicting Ventilating Noise, by L. L. Beranek, J. L. Reynolds and K. E. Wilson (*Journal Acoustical Society of America*, 25, March 1953, pp. 313–321).

[2] Noise of Centrifugal Fans, by L. L. Beranek, G. W. Kamperman and C. H. Allen (*Journal Acoustical Society of America*, 27, March 1955, pp. 217–219).

[3] *Acoustics*, by L. L. Beranek (McGraw-Hill Book Co., 1954, pp. 319–320).

[4] Output of a Sound Source in a Reverberation Chamber, by Richard V. Waterhouse (presented at Fall meeting of the *Acoustical Society of America*, 1955).

is appropriate to the type of mounting used. With these data the reverberant field and the direct sound field can be calculated for the equipment in any room in which it may be installed, provided the amount of acoustic absorption in the room is known.

A word of caution is in order at this point regarding the expression of the acoustic output of a noise source in terms of power. This means of expression gives a false sense of accuracy of measurements and a false sense of importance to small differences of sound power from one piece of equipment to another. Actually, differences in power less than a factor of 2 are insignificant in most cases since they result in a difference in sound pressure level of less than 3 db, which is barely perceptible by an untrained listener. Claims which might be made concerning the relative quietness of one equipment over another based on differences of 10 to 30 percent are insignificant but might well hoodwink the unsuspecting public if this information is improperly used.

It seems most appropriate to express sound power output of a source in terms of sound power level in decibels, like sound pressure level or intensity level, is expressed in logarithmic units*. Expression of the noise output of a source in terms of sound power level is a natural and extremely useful extension of the original concept of intensity level. The use of sound power level is discussed in detail in Reference 3, pp 314–319.

Noise Criteria: The demands of the consumer change. As he is able to obtain quieter devices, he demands them, and as his activities change, his requirements change also. For this reason noise reduction criteria change and the manufacturer must make continued improvements to meet them. There is no one spectrum which is best and no one criterion which can be applied. However, definite criteria have been determined where noise reduction is required to meet certain needs (Ref. 3, pp 416–429, and Ref. 5).

1. Damage risk criteria
2. Speech interference levels
3. Speech criteria
4. Loudness levels[6]

Damage risk criteria give an upper limit for the octave band sound pressure levels to which an average individual may be subjected during each working day over a normal working lifetime without risk of significant permanent hearing loss. Most of the noise levels with which we are interested here fall well below the damage risk level.

Where speech communication is important it has been demonstrated that the average of the sound pressure levels in the 3 octave bands 600–1200, 1200–2400 and 2400–4800 cps, determines the degree of speech interference. This average is called the speech interference level or *SIL*. Speech interference levels have been correlated with the types of communication carried on satisfactorily in offices, factories, broadcasting studios, etc. For example:

$SIL = 15$ for broadcasting studios
$SIL = 40$ for moderately quiet offices
$SIL = 65$ for factory working areas

A variation of the spectrum shape over a wide range does not greatly influence speech intelligibility when the *SIL* is maintained constant, but it is, nevertheless, a known fact that spectra which have much high frequency energy are unnatural and are generally considered annoying. Spectra normally encountered and, therefore, spectra which sound natural tend to have much greater intensity in the low frequency bands. A spectrum which has been found particularly suited to room acoustics and has been used

* Sound power level is usually defined as

$$L_W = 10 \log \frac{W}{10^{-13}} \text{ decibels re } 10^{-13} \text{ watts}$$

where

L_W = sound power level (often called *PWL*) in decibels re 10^{-13} watts
W = sound power in watts

[5] Psycho-physical Effects of Noise, by E. B. Newman (*Noise Control*, 1, July 1955, pp. 16–21).
[6] *Acoustic Measurements*, by L. L. Beranek (John Wiley & Sons, Inc., New York, 1949, pp. 524–526).

successfully by the discussers' firm in specifying noise reduction for a wide variety of installations is the so-called speech communication criterion or *SC* criterion. It is a spectrum having a relatively high level at low frequencies, with an initial slope of 10 db between the first and second band gradually tapering to a slope of 1 db between the 7th and 8th octave band. An *SC* curve is designated with a number equal to its speech interference level. This spectrum shape gives what is considered to be the maximum permissible noise in each frequency band for a specified speech interference level.

There are some noise reduction problems in which speech communication criteria are not sufficient in evaluating equipment noise. For example, an air-conditioning unit installed in an office may produce a noise below the speech communication criterion which would normally be recommended for such an office, but because the particular office may be unusually quiet, or because the spectrum shape of the noise produced by the unit is unnatural or contains single frequency components, it is objectionable. In such instances, it may be of value to resort to a calculation of loudness. Hardy and Bishop have presented one method of computing loudness. More recent data by S. S. Stevens[7] indicates that a somewhat different method of calculating total loudness gives a better fit to the subjective reactions of observers. The new data yield a new set of curves relating octave band sound pressure levels and loudness, and, further, in calculating the total loudness of all octave bands, it weights the loudness in the loudest octave band more heavily than that in the remaining bands, thus

$$S_t = S_m + F(\Sigma S - S_m)$$

S_t is loudness in sones

S_m is the loudness of the loudest octave band

ΣS is the sum of loudness of all octave bands

F is a factor that may vary between zero and one; for a large variety of conditions $F = 0.27$.

The calculation of loudness, although slightly more complicated than simple comparison of octave band sound pressure level measurements with sound level criteria, may yield the necessary information to determine the cause of subjectively annoying noise under special limited conditions. Tests of subjective reactions to noise at an Air Force Base[8] have indicated that total loudness level in phons gives slightly better correlation to the average individual reaction to noise than does speech interference level where speech communication is not a limiting factor.

It must be emphasized strongly that where the sound pressure level is not constant from one position to another, loudness and loudness level cannot be extrapolated from one position to another. This fact has been implied by the authors in their statement that "the conversion from physical to subjective units is never made unless one can conceive of a person actually listening at the point. . . . calculation of the loudness inside a ventilating duct would be ridiculous." In like manner, it is ridiculous to calculate loudness of a machine or other equipment at some standard distance if it is to be heard from some other distance. It is not even possible to compare the loudness of 2 different machines under a standard condition if they are to be heard under another condition. Loudness can be calculated only when the sound pressure level in each octave band is known at the point where the sound is to be heard.

Rating the sound output of a machine in terms of its loudness at some standard distance or in some standard reverberation room does not specify its spectrum and, therefore, does not give sufficient information to recalculate its subjective loudness under its intended operating conditions. It is not possible to rank various pieces of machinery by a single-number loudness or loudness level, and for this reason it is not possible to write general standards using this notation. To do so is little better than to write

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[7] Sones, Spectra, and Summation, by S. S. Stevens (presented at the Fall Meeting of the *Acoustica Society of America*, 1955).

[8] Subjective Noise Ratings in Air Force Base Offices, by L. L. Beranek (presented at the Fall Meeting of the *Acoustical Society of America*, 1955).

specifications around the single-number, overall sound pressure level which has already been shown to be inadequate.

In reviewing criteria we should remember that the final and often the most exacting criterion for the air-conditioning engineer is consumer acceptance. Frequently after other criteria, such as speech interference level, speech communication criteria, or maximum loudness level have been met, the consumer will buy the quietest device which meets his needs and is competitively priced. Therefore, it is advisable for the manufacturer to compare his product objectively with competitive units to determine where and to what extent further reduction in noise might result in profitable competitive advantage. Octave band analysis and comparison of sound pressure levels between the best and poorer equipment will generally determine the troublesome frequencies and indicate the type of noise reducton needed.

Noise measurement and its specification is not a simple matter. It is, nevertheless, an important, practical, economic consideration for the heating, ventilating and air-conditioning engineer. Although it is difficult for those not trained in the field, it is profitable in terms of dollar income to learn the more detailed methods of handling acoustical problems. Simplified methods, although apparently easier to understand on first reading, tend to be inadequate and give a false sense of security which may only confuse the issue.

Our recommendation is that the sound power or sound power level be determined for each of 8 octave frequency bands. We also recommend that directivity patterns be determined at several frequencies spaced over the frequency scale as appropriate to the device being tested (usually 2 or 3 directivity patterns will suffice). From this information sound levels in ducts or rooms of any known configuration and liveness can be calculated, either in the near field of the sound source or in the reverberant field. The effect of addition of transmission loss or absorptive material, or the effect of an increase in distance can be calculated directly.

Summary: 1. We strongly support the authors in urging the adoption of noise standards within the Society on the basis of octave band levels to replace the relatively useless overall noise levels.

2. Noise sources in general require for their description a determination of both sound power in octave bands and directivity patterns. Such measurements need to be carried out with the noise source in an environment as nearly as possible like that in which it will be used.

3. Criteria for noise reduction and control must be chosen with regard for the type of activity of interest. Not one but several criteria exist which apply for special types of noise control.

4. The loudness or loudness level concept offers little simplification in rating the *noisiness* of machinery, and it may lead to a false sense of security by neglecting information needed to give a complete picture. The calculation of loudness level at any particular distance or under any particular room condition has very limited utility, and cannot be used for writing general standards for equipment noise. Furthermore, a single loudness or loudness level number offers little aid to the heating, ventilating and air-conditioning engineer in the detailed design of a quieted building.

5. It is our recommendation that specifications for noise control be left in terms of physical measurements. The user will be able from these to calculate *SC* ratings, loudness, loudness level or any other quantity of significance for his particular application.

R. V. WATERHOUSE*, Washington, D. C. (WRITTEN): This paper deals with a topic which is of considerable practical importance at this time. The authors outline a method of computing sound and loudness levels in rooms when sound sources are present. The method assumes the sound energy emitted by the source is uniformly distributed throughout the room.

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* National Bureau of Standards.

Now without some kind of simplifying assumption such as this the calculation of noise levels in rooms would be impossibly complicated. However, the question does arise as to how large an error in the results this assumption leads to.

The question is of practical importance; if a specification requires the level at a point in a room to be less than 50 db, the acoustical engineer must know what safety factor to use in his design.

The uniformity of sound distribution in a room depends on the amount of absorption in the room and the spectrum of the source, and it is probably difficult to make an estimate of the probable errors in the computed sound levels due to these factors.

Thus I think some experimental data showing predicted versus measured levels, under different source and room conditions, would be most useful in establishing the accuracy to be expected from the method.

C. M. ASHLEY, Syracuse, N. Y. (WRITTEN): The authors have presented in very clear terms the method of evaluating the noise in a room resulting from equipment noise.

Their emphasis on the importance of the use of the sound power concept on a frequency band basis needs constant reiteration.

It is unfortunate that in describing the method of measurement they have limited themselves to only one method and have *not* indicated some of the difficulties of the method, which are due primarily to the presence of standing waves in a reflective room. A combination of this method at low frequencies with the reverberant room method at high frequencies might prove to be superior to either, used alone.

It is believed that for many types of apparatus, a test under free field or modified free field conditions is to be preferred. We are now using a method of testing which involves a reflective floor and wall in an otherwise absorptive room with good success. Another method which is desirable for some equipment is to test in a modified reverberant room simulating conditions of actual use and using a calibrated source.

Both of these methods have the advantage of taking into account the directivity of sound, which the reverberant method cannot do.

For the testing of fans which are to be connected to ductwork, it is believed that the most satisfactory procedure is a test in the duct, itself, and using an absorptive terminal, since this gets inside the reflective effects which occur at the duct termination.

A combination of this method at low frequency with the reverberant method at high frequency might be better than either alone.

I seriously question the desirability of using the loudness concept as described by the authors as a means of evaluation, for the following reasons:

(1) Loudness as such is not a true measure of the annoyance of a noise. Annoyance is also affected by such factors as the frequency distribution, presence of discrete frequency components and time variations. This is particularly true at low noise levels.

(2) The method correlates with subjective loudness only for certain types of noises. For other frequency distributions the correlation is unacceptable.

(3) Recent studies have indicated a different equation of correlation between frequency distribution and loudness.

(4) We believe that there are other available criteria which give more satisfactory correlation.

I would like to emphasize the importance of what the authors have said about the need for further study of sound sources and attenuation factors in air conditioning systems. It is to be hoped that the current research project being carried at the laboratory of the Society will supply much of this needed information.

H. A. LOCKHART, Morton Grove, Ill.: We believe that the authors have ably surveyed the existing art of noise measurement and have pointed out the specific direction in which research should be undertaken.

It might be well at this point to report on the activity of our Sound and Vibration Control Committee. During the past several years this Committee has gradually formed a consistent program for the direction of research effort on the broad subject of

sound and vibration. Perhaps one of the most important steps has been the liaison between the National Association of Fan Manufacturers and the ASA.

In 1953, jointly with NAFM, we outlined a program, which is a long-range plan of the Committee, as follows: Projects of immediate importance:

1. A method for predicting noise level in conditioned space. Development of qualitative and quantitative information on the transmission of sound from equipment to the spaces being considered.
2. Determination of the preferred technique of measurement of sound power level on a frequency band basis for fans and other air-handling equipment.
3. The determination of the attenuation characteristics of duct systems both natural and due to the use of duct lining sound filters, plenums and other attenuating devices.
4. The generation of noise in duct systems due to the flow of air in the duct through branch take-offs, sound around elbows and through dampers, grilles, and other devices.
5. Criteria for determining the annoyance to customers of noise of the type generated by air-conditioning equipment.
6. The further development of instrumentation and techniques for sound measurement.
7. The study of the acoustic absorption properties of typical rooms or spaces in which air-conditioning equipment or outlets are located.
8. The development of a standardized method for predicting noises generated and propagated by air-conditioning systems both acoustically treated and untreated.

Work on all of these 8 items should cover air velocity in both the conventional and the high-range.

We have several projects of secondary importance which I will not report on at this time. A definite research project on fan noise was recommended to the Research Committee, and was approved and activated at the Laboratory at Cleveland. Progress on this project has been rapid. I think, we have all been surprised at how very well this has been going.

We have received approval to start another project on the attenuation of noise in duct systems in the near future.

We believe you will agree that the program outlined is an ambitious one. We hope you will go along that this program should receive our full support. We are of the opinion that the work which is contemplated will add materially to the background of our engineering knowledge and will help us all in the Society.

This isn't an engineering comment, but the work is going to depend on the financial backing which the Society provides. I hope that this work on sound means enough to all of us so that we are going to be able to carry forward this program as outlined over the coming years to a successful conclusion.

AUTHORS' CLOSURE (Dr. Hardy): We started out with what I hoped was a simple discussion and it got a little deeper as the discussion went along. In the 15 minutes before me you can be assured that I can't present all the ramifications of sound engineering. What I was attempting to do was to present the first chapter. Some of the comments seem to be on the third and fourth chapters.

One of the speakers seemed to make quite a point about directivity. I don't believe it is necessary in this Society to point out that directivity is important. If you dump a lot of air into a room, you have also got to pay some attention to where you point it. Just as it is important not to spray air on the tops of people's heads, it is important not to spray sound back of their ears.

It turns out in most engineering problems that the same distance that one would spray air indoors is the same distance that direct sound can be sprayed from a sound source in the presence of reverberant sound. There isn't any rule for this, but it usually turns out to be approximately that far. So generally if you are beaming air away, you will be beaming sound away.

I do believe directivity is fairly important. It gets to be more important in air-conditioning units and some appliances than it does in large ventilating duct systems. I think, however, that a discussion of it should be postponed to the third chapter.

There was another comment made about the accuracy of this system. Well, it depends on the accuracy somewhat of the experimenters who work with it. With proper care one can calculate these things out to one or two decimals and that is quite sufficient for nearly all the problems in air conditioning engineering.

Actually, this system of measurements isn't a new proposal that we are presenting here. Some of the commentors seem to be alarmed that there was something altogether new about it. It is being used and has been used for a long time in other fields of engineering. There are something like a dozen industries now who are using reverberant rooms for making the measurements, not all in the air-conditioning industry. Some of them have very accurate calculation laboratories.

Also, this technique has been used in the armed forces and is the method now used in the navy for evaluating the ventilating equipment on board ships. It is useful. So I am sure that it will come into more practice in the future.

1565

No. 1565

ELECTRIC ANALOGUE STUDIES OF SINGLE WALLS

By HARRY BUCHBERG*, LOS ANGELES, CALIF.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with the Department of Engineering, University of California, Los Angeles.

IN THERMAL CIRCUIT representation the error introduced by lumping the properties of the thermal system is a function of the fineness of the lumping, the pattern of lumping, the boundary conditions, the length of time after a change in input occurs, and the location in the network where the value of temperature or heat flux is required. Some attempts have been made to evaluate the lumping error of particular resistance-capacitance networks having simple boundary conditions using experimental[1] and analytical[2] methods. Reference 3 presents an approximate method of lumping based on filter network analysis, but further verification appears to be necessary. No attempt at generalization is available in the literature. Lacking a good analytical solution to the problem of lumping and because of the need to study different lumping arrangements in connection with the thermal circuit representation of a test house (see Reference 4), it was desirable to investigate by electric analogue solution several thermal circuit configurations of a single wall section.

The wall investigated is of Douglas fir frame construction, consisting of 1-in. exterior sheathing, 2- × 4-in. vertical studs placed 16 in. on center, and ¼-in. interior plywood. The studies include both an uninsulated wall and a wall insulated with standard 3-in. rockwool batts. In addition, the effects of different values of outside and inside surface conductance on surface temperature and heat flux were determined.

Electric analogue solutions of several different thermal circuit configurations, shown in Fig. 1, having identical boundary conditions are presented. The results of these solutions are given in Figs. 2 through 15 in which outside and inside surface temperatures and heat flux across the inside surface boundary are plotted as a function of time. The solar input and outside air potential were identical to those described for the west wall in the test house circuit given in Reference 4, while the

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[1] Exponent numerals refer to References.

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inside air potential was held constant and equal to 72.5 F. The inside and outside surface-to-air resistances were based on combined radiation and convection conductances.

SYMBOLS

= source of constant temperature, or potential

= source of periodically varying temperature, or potential

= heat or current source

= thermal resistance

= thermal capacitance

= ground or reference potential

The input devices used to represent the diurnal variation in air temperature and solar energy absorbed by the exposed surface are described in Reference 4. Potential measurements for the solutions were made by means of a multi-channel recording oscillograph, d-c amplifier and cathode-follower inputs.

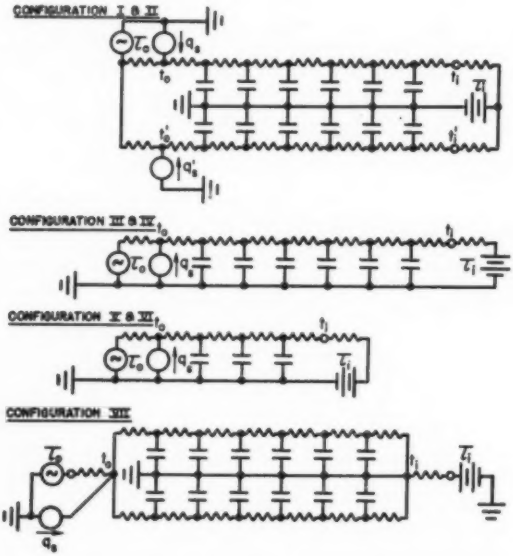

FIG. 1—THERMAL CIRCUITS REPRESENTING A WOOD FRAME WALL SECTION

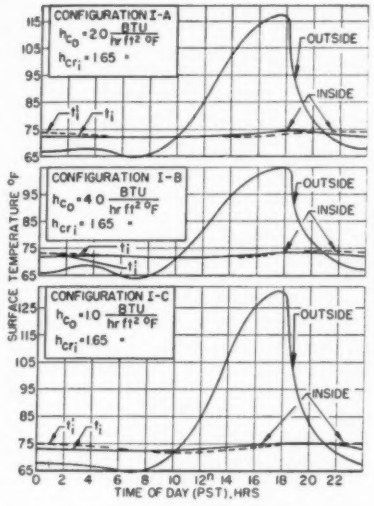

FIG. 2—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT VALUES OF OUTSIDE CONDUCTANCE AS NOTED (INSULATED WALL)

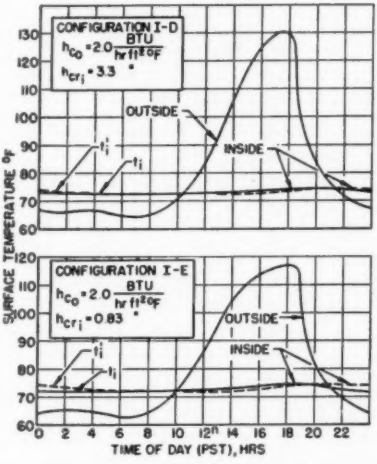

FIG. 3—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT VALUES OF INSIDE CONDUCTANCE AS NOTED (INSULATED WALL)

CIRCUIT CONFIGURATIONS

The methods of lumping, in the configurations studied (see Fig. 1), may be summarized as follows:

Circuit a — configuration I and configuration II: Configuration I and configuration II are identical except for the value of conduction path resistance and capacitance. The values of circuit parameters in I are based on an insulated wall and in II on an uninsulated wall or free air space between studs. This circuit represents the parallel conduction path through the space and studs. Each material in the composite structure was lumped separately in equal parts; 2 lumps for the exterior sheathing, 3 lumps for the space, 1 lump for the interior plywood sheet, 6 lumps for the path through solid wood. This lumping was considered to be the most accurate representation and all other configurations were compared with it.

Circuit b — configuration III and configuration IV: Configuration III and configuration IV are identical except that III represents the insulated wall and IV the uninsulated wall. This is a simplified circuit consisting of only a single conduction path. There are several ways in which the circuit parameters might be determined: the resistance and capacitance of the studs could be lumped with the space, the stud area could be treated as additional air space or insulation, or the stud area neglected entirely on the basis that the heat flow through that leg of the circuit is small compared to flow through the space. For steady state, the last course would look reasonable for the uninsulated wall, but not for the insulated wall. Actually,

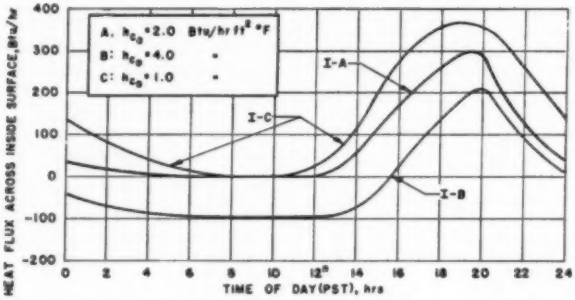

FIG. 4—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR AN INSIDE CONDUCTANCE OF 1.65 BTU/(HR) (SQ FT) (F DEG) (INSULATED WALL)

with a periodic input the heat flow in from the stud area is significant, compared to the air space, during the night hours because of the capacitive effect. The difficulty with the first course is that the capacitive effect of the studs is weighted much too highly, introducing a significant error in the phase difference between the outside and inside surface temperature cycle. The second course was, therefore, chosen and might be expected to yield slightly high values of heat flux in during the day and out during the night.

Circuit c — configuration V and configuration VI: Configuration V and configuration VI are identical in lumping and the same as III and IV, respectively, except that the number of lumps is reduced. In this case each material in the composite wall is represented by one lump.

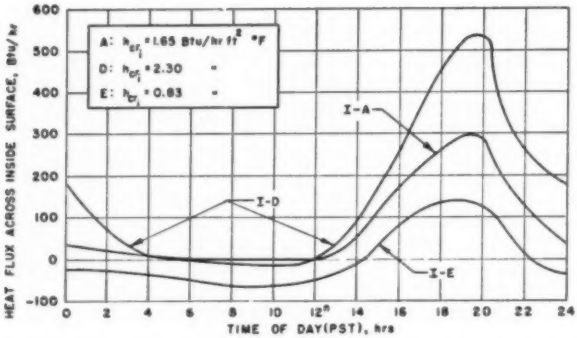

FIG. 5—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR AN OUTSIDE CONDUCTANCE OF 2.0 BTU/(HR) (SQ FT) (F DEG) (INSULATED WALL). NOTE: THE FIGURE 2.30 SHOULD READ 3.30

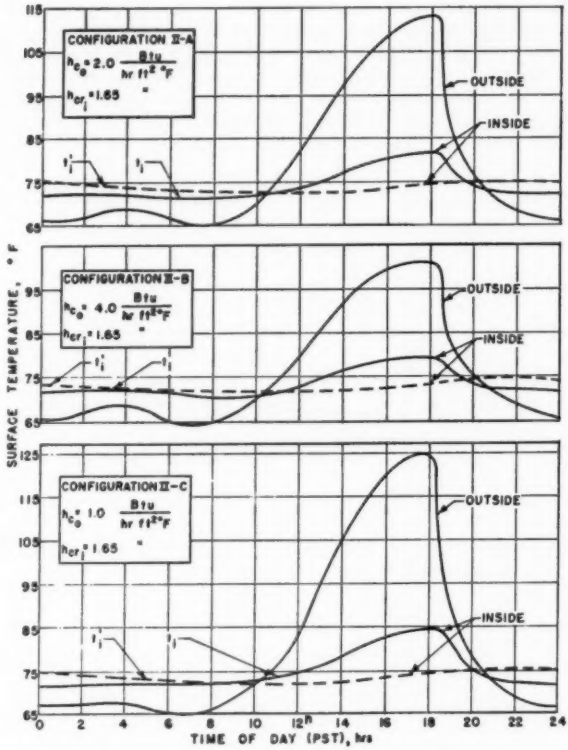

FIG. 6—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT VALUES OF OUTSIDE CONDUCTANCE AS NOTED (UNINSULATED WALL)

Circuit d — configuration VII: This configuration represents an uninsulated wall accounting for the parallel flow path through the studs and space but assuming a common potential at the inside and outside surface for both paths. This configuration has the desirable advantage over II in that it reduced the number of boundary inputs required in the circuit.

Appendix A presents the values of thermal resistance and capacitance used in the different circuit configurations and a tabulation of the thermal-electrical scale factors.

CIRCUIT SOLUTIONS

The graphs presenting the various circuit configuration solutions are grouped to illustrate the influence of several factors.

Figs. 10 *and* 11 illustrate the effect of fineness and arrangement of lumping on the outside and inside surface temperatures and on the heat flux across the inside surface boundary for the insulated west wall. As expected, the simplified configuration resulted in peak outside surface temperatures slightly lower than the basic configuration I-A, and peak inside surface temperatures slightly higher. The total heat flux predicted by the simplified lumping configuration III-A is somewhat greater than for the basic lumping, but the load predicted with a reduced number

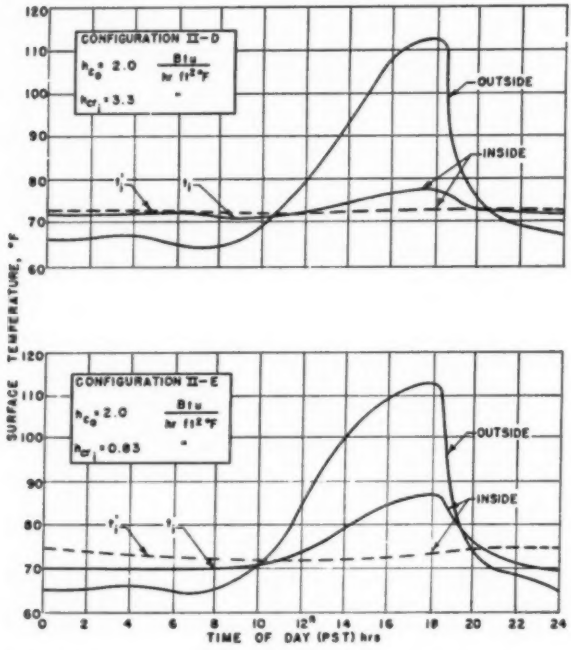

FIG. 7—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT VALUES OF INSIDE CONDUCTANCE AS NOTED (UNINSULATED WALL)

of lumps is less. The coarser lumping also tended to reduce slightly the phase lag through the wall.

Figs. 12 *and* 13 illustrate the effect of fineness and arrangement of lumping on the outside and inside surface temperatures and on the heat flux across the inside surface boundary for the uninsulated west wall. In this case the effect of considering the stud area as space is less pronounced than for the insulated wall. However, a slight decrease in outside surface temperature and a slight increase in heat flux during the day are apparent. A larger difference occurs during the night when the stud capacitance appears to aid considerably the reduction in losses at night. This

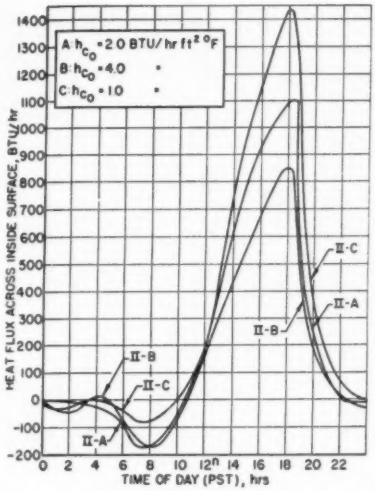

FIG. 8—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR AN INSIDE CONDUCTANCE OF 1.65 BTU/(HR) (SQ FT) (F DEG) (UNINSULATED WALL)

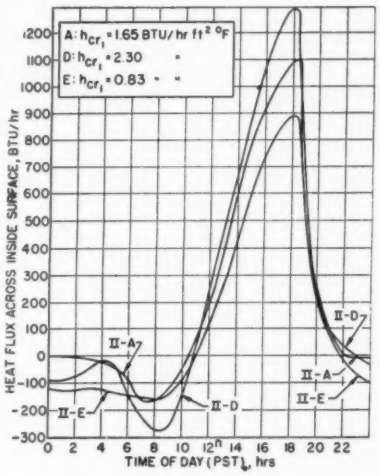

FIG. 9—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR AN OUTSIDE CONDUCTANCE OF 2.0 BTU/(HR) (SQ FT) (F DEG) (UNINSULATED WALL)

is not predicted by the simplified configuration IV-A. The reduction in the number of lumps in configuration VI-A does not seem to affect the results greatly. The heat flux prediction is only slightly larger.

Figs. 14 *and* 15 compare the surface temperature and heat flux predictions of 3 different circuit arrangements with a relatively low value of inside surface conductance. The larger inside surface-to-air resistance had the effect of minimizing the difference between the basic configuration II-E and the simplified circuit IV-E, particularly during the night hours. The heat fluxes predicted during the afternoon by the simplified circuit are still slightly high; however, the predictions, in general, compare well with the basic circuit. Predictions of heat flux based on circuit configuration VII-E compare well with the basic circuit except during the night and early morning hours when the capacitive effect of the studs is exaggerated. This may be expected because of the short circuit of the parallel heat conduction paths at the inside surface.

Figs. 2 *through* 9 illustrate the effect of different values of inside and outside surface conductance on the surface temperatures and heat flux across the inside surface. These effects are shown for both the insulated and uninsulated west wall. These results indicate the importance of the predictions of unit thermal conductance in estimating the cooling loads and apply only to the wood frame construction represented in these studies.

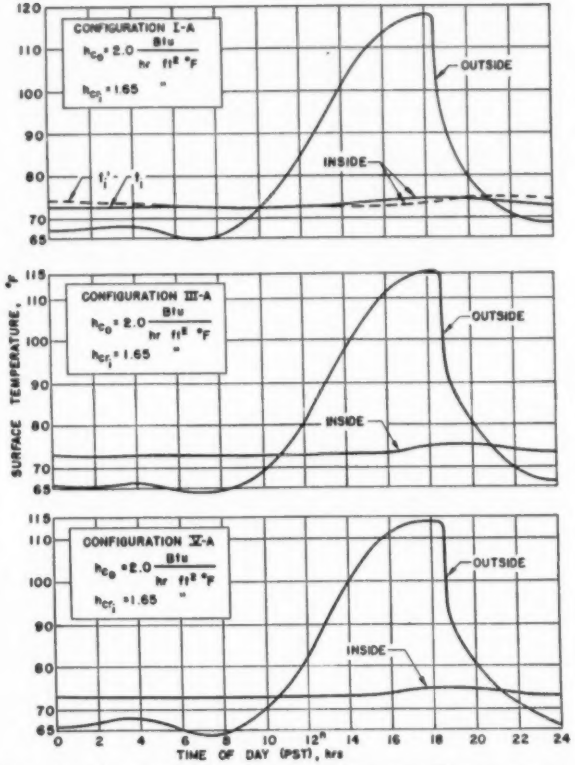

FIG. 10—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE USING DIFFERENT LUMPING CONFIGURATIONS AS NOTED (INSULATED WALL)

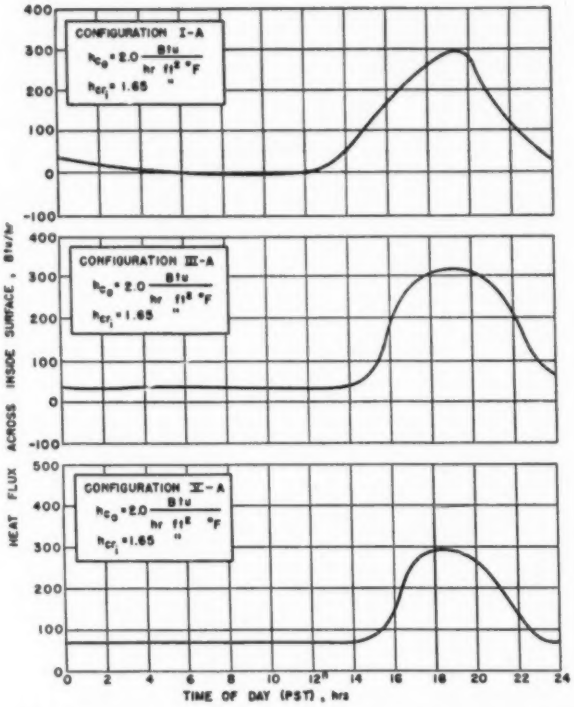
CONFIGURATION I-A
h_{c_o} = 2.0 Btu/hr ft² °F
h_{cr_i} = 1.65 "
CONFIGURATION III-A
h_{c_o} = 2.0 Btu/hr ft² °F
h_{cr_i} = 1.65 "
CONFIGURATION V-A
h_{c_o} = 2.0 Btu/hr ft² °F
h_{cr_i} = 1.65 "
HEAT FLUX ACROSS INSIDE SURFACE, Btu/hr
TIME OF DAY (PST), hrs

FIG. 11—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT LUMPING CONFIGURATIONS (INSULATED WALL)

ING CONFIGURATIONS AS NOTED (INSULATED WALL)

LUMPING CONFIGURATIONS (INSULATED WALL)

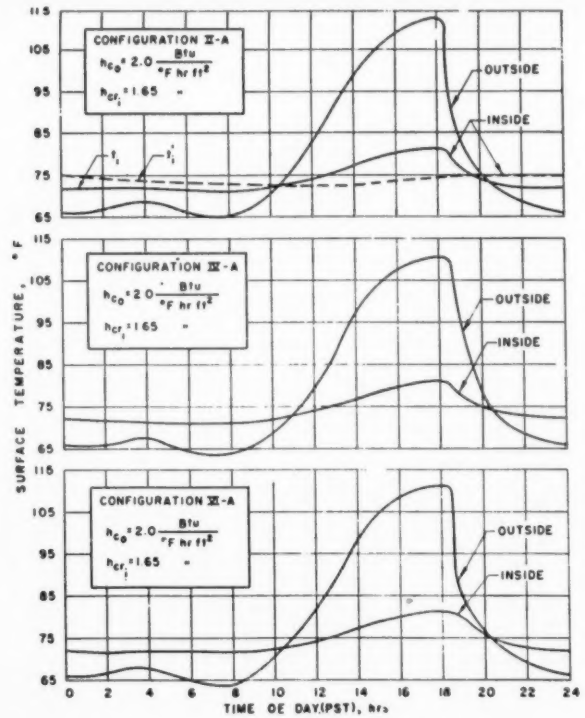

FIG. 12—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE USING DIFFERENT LUMPING CONFIGURATIONS AS NOTED (UNINSULATED WALL)

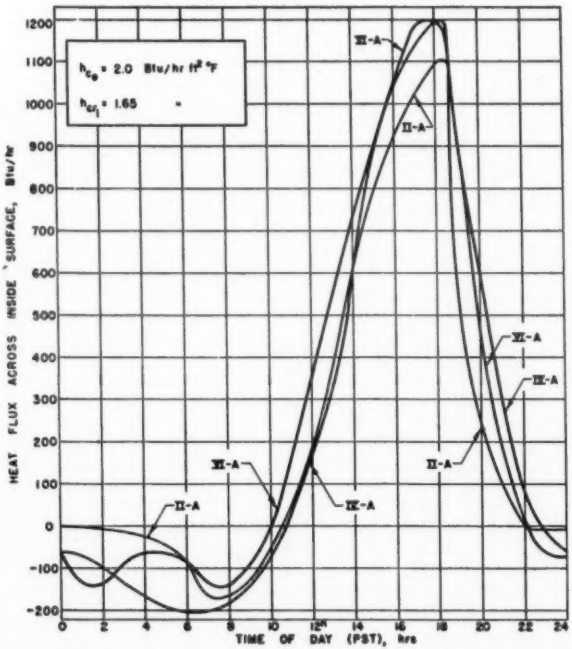

FIG. 13—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT LUMPING CONFIGURATIONS (UNINSULATED WALL)

REFERENCES

1. Accuracy of Measurement in Lumped R-C Cable Circuits As Used in the Study of Transient Heat Flow, by Victor Paschkis and M. P. Heisler (*Transactions of the American Institute of Electrical Engineers*, Vol. 63, April 1944, p. 165).

2. Limits of Accuracy of Electrical Analog Circuits Used in the Solution of Transient Heat Conduction Problems, by E. O. P. Klein, Y. S. Touloukian and T. R. Eaton (ASME Paper No. 52-A-65, presented at Annual Meeting, New York, N. Y., November 30–December 5, 1952).

3. ASHVE RESEARCH REPORT No. 1497—Circuit Analysis Applied to Load Estimating, Appendix C, by H. B. Nottage and G. V. Parmelee (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 59).

4. ASHAE RESEARCH REPORT No. 1543—Electric Analogue Prediction of the Thermal Behavior of an Inhabitable Enclosure, by Harry Buchberg (ASHAE TRANSAC-

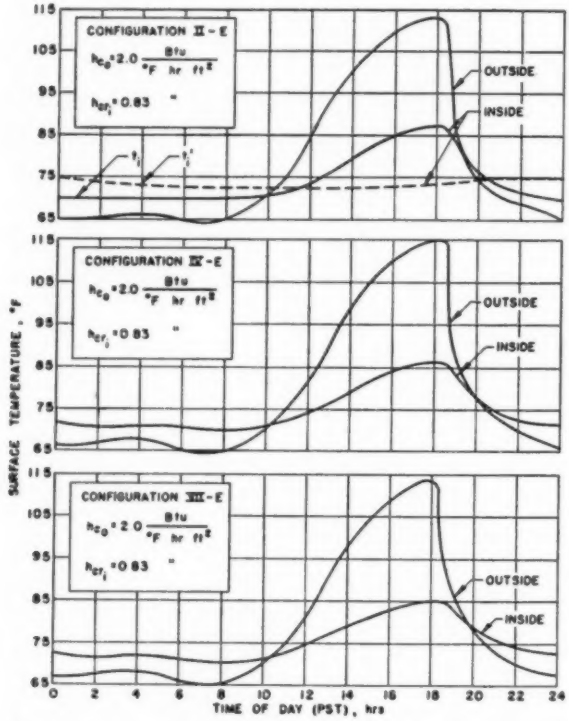

FIG. 14—DIURNAL VARIATION OF SURFACE TEMPERATURES COMPUTED WITH ELECTRIC ANALOGUE USING DIFFERENT LUMPING CONFIGURATIONS AS NOTED (UNINSULATED WALL)

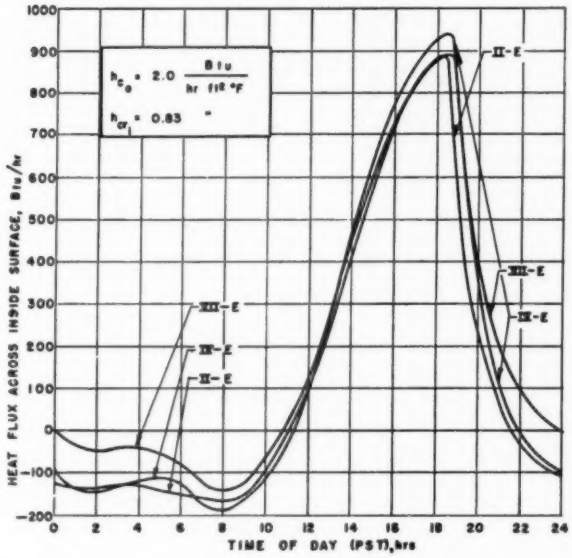

FIG. 15—DIURNAL VARIATION IN HEAT FLUX ACROSS INSIDE SURFACE COMPUTED WITH ELECTRIC ANALOGUE FOR DIFFERENT LUMPING CONFIGURATIONS (UNINSULATED WALL)

TIONS, Vol. 61, 1955, p. 339), also, Report No. 54-84, Department of Engineering, University of California, Los Angeles, September 1954.

APPENDIX

Additional structural details, values of thermal resistance and capacitance for the circuits shown in Fig. 1, and the thermal-electrical scale factors used are presented in Fig. A-1 and Tables A-1, A-2, and A-3. It should be noted that Table A-1 presents the values of thermal resistance for configurations I-A, II-A, III-A, IV-A, V-A, VI-A, and VII-E. Configurations I-B, C, D, and E; II-B, C, D, and E; and IV-E differ from the basic configurations in Table A-1 only in the values of the inside and outside surface-to-air resistance. The values of h_{co} and h_{cri} are given on the graphs in Figs. 2 through 15; the corresponding values of resistance are given in Table A-1. The value of h_{ro}, equivalent outside radiation conductance, was constant for all runs and equal to 0.917 Btu/(hr) (sq ft) (F deg).

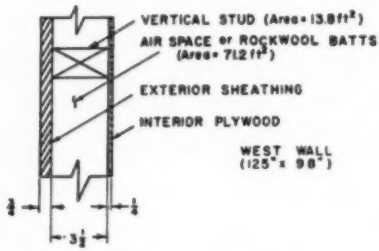

FIG. A-1—WEST WALL STRUCTURAL DETAILS

TABLE A-1. VALUES FOR RESISTANCES OF CIRCUITS SHOWN IN FIG. 1

[Going from left to right, (Fahrenheit degree) (hour) per Btu]

Configuration I-A	1	2	3	4	5	6	7	8	9
Top Row (Insulation Path)	0.00481	0.00338	0.00675	0.0306	0.0345	0.0345	0.0295	0.00225	0.0085
Bottom Row (Stud Path)	0.0248	0.0348	0.0697	0.0697	0.0697	0.0697	0.0697	0.0348	0.0440
Configuration II-A	1	2	3	4	5	6	7	8	9
Top Row (Air Space Path)	0.00481	0.00338	0.00675	0.00533	0.00390	0.00390	0.00420	0.00225	0.0085
Bottom Row (Stud Path)	0.0248	0.0348	0.0697	0.0697	0.0697	0.0697	0.0697	0.0348	0.0440
Configuration III-A	1	2	3	4	5	6	7	8	9
	0.00403	0.00283	0.00565	0.0256	0.0456	0.0456	0.0247	0.00189	0.00712
Configuration IV-A	1	2	3	4	5	6	7	8	9
	0.00403	0.00283	0.00565	0.00446	0.00327	0.00327	0.00352	0.00189	0.00712
Configuration V-A	1	2	3	4	5	6			
	0.00403	0.00565	0.0742	0.0704	0.00189	0.00712			
Configuration VI-A	1	2	3	4	5	6			
	0.00403	0.00565	0.01055	0.00679	0.00189	0.00712			
Configuration VII-E	1	2	3	4	5	6	7	8	9
Top Row	0.00403	0.00338	0.00675	0.00533	0.00390	0.00390	0.00420	0.00225	0.01418
Bottom Row		0.0348	0.0697	0.0697	0.0697	0.0697	0.0697	0.0348	

Configuration	Surface to Air Resistance, (Fahrenheit degree) (hour) per Btu			
	1 (Top)	1′ (Bottom)	9 (Top)	9′ (Bottom)
I and II -A	0.00481	0.0248	0.00850	0.0440
-B	0.00288	0.0149	0.00850	0.0440
-C	0.00735	0.0379	0.00850	0.0440
-D	0.00481	0.0248	0.00425	0.0220
-E	0.00481	0.0248	0.01692	0.0874
IV-A	0.00403		0.00712	
IV-E	0.00403		0.0142	

TABLE A-2. VALUES FOR CAPACITANCES OF CIRCUITS SHOWN IN FIG. 1
(Going from left to right, Btu per Fahrenheit degree)

CONFIGURATION	EXTERIOR SHEATHING					INTERIOR PLYWOOD
			Insulation			
I Top Row (Insulation Path)	37.8	37.8	13.9	13.9	13.9	20.1
			Air Space			
II & VII Top Row (Air Space Path)	37.8	37.8	0.118	0.118	0.118	20.1
			Insulation			
III	45.1	45.1	16.6	16.6	16.6	30.0
			Air Space			
IV	45.1	45.1	0.141	0.141	0.141	30.0
			Insulation			
V		90.2		49.8		30.0
			Air Space			
VI		90.2		0.423		30.0
			Solid Path of Douglas Fir			
I, II, & VII Bottom Row (Stud Path)	14.7	14.7	14.7	14.7	14.7	14.7

TABLE A-3. SUMMARY OF RATIOS AND UNITS OF ANALOGOUS ELECTRICAL AND THERMAL QUALITIES

QUANTITY	UNITS		SCALE FACTORS	
	THERMAL	ELECTRICAL	RATIO	VALUE
Time	hrs	sec		2
Capacity	$\frac{\text{Btu}}{\text{F deg}}$	Farads	$\frac{C_t}{C_e}$	8×10^6
Resistance	$\frac{\text{F deg}}{\text{(Btu/hr)}}$	Ohms	$\frac{R_e}{R_t}$	16×10^6
Potential	F deg	Volts	$\frac{E}{t - t^*}$	1
Rate of Energy Transfer	$\frac{\text{Btu}}{\text{hr}}$	$\frac{\text{Coulombs}}{\text{sec}}$ or Amperes	$\frac{q}{i}$	16×10^6

t^* = Reference temperature. Subscripts: t = thermal circuit element; e = electrical circuit element.

NOMENCLATURE*

h_{co} = unit thermal conductance at outside surface, Btu per (hour) (square foot) (Fahrenheit degree).
h_{cri} = equivalent unit thermal conductance at inside surface, Btu per (hour) (square foot) (Fahrenheit degree).
q_s = solar input, or solar energy absorbed by the exposed surface, Btu per hour.
t_i = inside surface temperature, Fahrenheit degrees.
t_o = outside surface temperature, Fahrenheit degrees.
τ_i = inside ambient air temperature, Fahrenheit degrees.
τ_o = outside ambient air temperature, Fahrenheit degrees.

DISCUSSION

N. R. GAY, Ithaca, N. Y. (WRITTEN): This paper, together with the earlier work by the same author, represents a worthwhile addition to the information being made available to the membership on methods of analyzing periodic heat flow. It would seem to me that it should be of interest to engineers to compare results of studies such as this with presently used methods of calculation and prediction. Having only reproductions of the curves presented in the paper at the June 1955 meeting the accuracy of resultant calculations is necessarily limited.

If one were to take the solar energy absorptions and diurnal variation in outdoor air temperature given in the earlier paper it would be possible to calculate a sol-air temperature to be used in the study under discussion. Periodic heat flow values from THE GUIDE are based on an outside film conductance of 4.0 Btu per (hr) (sq ft) (F deg) and an inside film conductance of 1.65 Btu per (hr) (sq ft) (F deg). Thus, they are comparable to the B series in this paper except for the fact that the outside conductance is really 4.917 Btu per (hr) (sq ft) (F deg) as built into the electrical analogue to include effects of low-temperature radiation. Perhaps a better procedure for comparison would be to average the results of the A and B series which would give an indicated conductance of 3.917 Btu per (hr) (sq ft) (F deg) for the outside, all other things remaining the same.

At any rate, configuration I from THE GUIDE would indicate a time lag of approximately 3.4 hours for the insulated wall. This checks fairly well as the maximum heat flux occurs at about 7:30 pm and the sol-air temperature I calculate to be a maximum at about 4 pm from the figures given in the earlier paper. Using an amplitude decrement factor of 0.68 gives a heat flux of 194 Btu per hr as compared to a figure of about 200 Btu per hr from Curve I–B, Fig 4. If curves I–A and I–B are averaged, however, Fig. 4 indicates a value of about 250 Btu per hr which is considerably greater than the value predicted by THE GUIDE. This is contrary to the results of some other investigators who generally consider THE GUIDE predictions to be on the high side.

Similarly for Configuration II series B, THE GUIDE would indicate a time lag of about 2 hours for this composite structure without insulation, and with a decrement factor of 0.87 would yield a maximum heat flux of about 840 Btu per hr. Again this is lower than the average of A and B, but quite comparable to B alone.

In general, this paper would indicate that considerably more time should be devoted to the establishment of reasonable values of thermal conductances since analogue results, of whatever type, can only be as useful as the component data built into them are realistic.

One very small point needing correction is that Fig. 5 is plotted for an *outside* conductance of 2.0 not *inside* as stated in the labeling in the original paper.

L. W. NELSON, Hopkins, Minn. (WRITTEN): The author has contributed some very interesting and valuable basic information to the field of thermal circuitry. Certainly information of this type is essential in evaluating the results of a thermal circuit applied

* The primed quantities refer to the path through the studs.

to a transient load. Because we do not have at the present time an exact method for duplicating distributive networks by resistance and capacitance, the question always arises as to the most accurate method of representing the conversion from a distributive thermal system to a lumped electrical system and the resulting error in the assumed circuit.

It would be most interesting to see the results of the different configurations compared with test results of an actual wall section of similar construction and subjected to the same transient load. It would also be interesting to compare the different configurations to an electrical analogue of the following type.

This method consists essentially of dividing the wall section into a number of small areas with the mass per unit depth assumed concentrated at the center of each. By

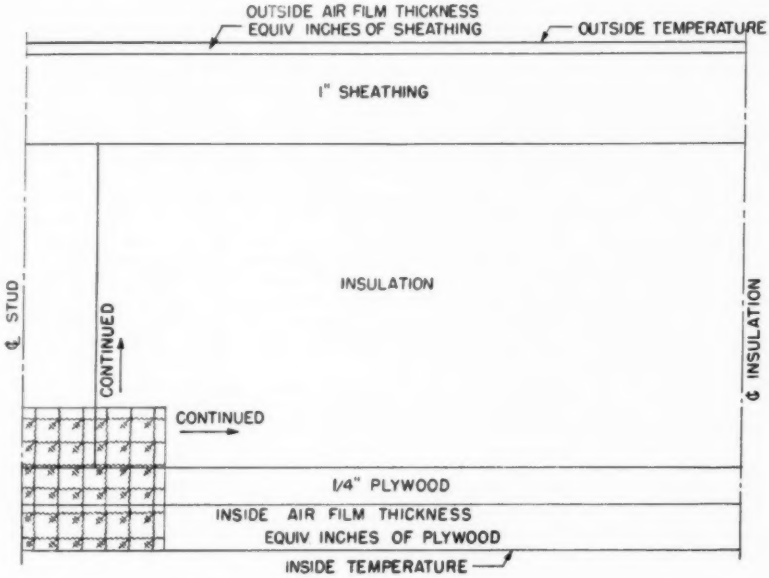

FIG. A—ELECTRIC ANALOGUE OF WALL SECTION

building a scale model of the wall section, an equivalent electrical capacitance may be inserted for each concentrated mass. Each area may then be connected by an equivalent electrical resistance depending upon the thermal resistance of the material in question. A transient input could be applied to the outside temperature and the heat flux and inside surface temperatures could be measured and computed for various locations within the wall. Under steady state this analogue would give the same results as using the *relaxation method.*

The accuracy of a method of this type compared to an actual wall section would depend upon the number of areas used and the knowledge of the thermal properties of the wall section. The results, however, should be closer to an actual wall section than that obtained with configurations normally used in thermal circuits.

I would like to have the author's comment on using an electric analogue of the type mentioned, not only to show the deviations of the simplified circuits but also the deviations of Configurations I and II.

In the analog that we have built at Minneapolis-Honeywell we have chosen to represent the wall section as having isothermal planes on both the interior and exterior surfaces of the stud with parallel heat flow through the studs and insulation or air space as the case may be. It would be very interesting to us to see how this method would compare to the configurations the author has shown.

H. B. NOTTAGE, Encino, Calif. (WRITTEN): This paper demonstrates a basic consideration in thermal circuit analysis. While the solution presented is instructive especially in regard to the author's previous paper, no generalization is made because, indeed, this cannot yet be done.

It remains my firm opinion that the most promising approach to general rules, applicable to a definable range of air-conditioning problems, is by way of circuit analysis. The uni-dimensional passive circuit in the form of a *ladder* of so-called T-elements is basic to lumped-parameter thermal circuits, and I strongly recommend that ASHAE and the University of California should undertake to generalize a lumping rule, as far as may be reasonable, for this sample case with representative boundary conditions as a simple starting point.

S. F. GILMAN, Syracuse, N. Y. (WRITTEN): I note several points in this paper which cause me concern. When the different configurations for, say, the insulated wall are subjected to identical boundary conditions, then the steady state heat flow for each should be the same. Hence, the total amount of heat removed over each 24-hr period should also be the same. Scrutiny of Fig. 11 indicates the total amount of heat, as represented by the areas under the curves, is not the same for all 3 configurations. Planimetering of these areas yields about 1800 Btu for I-A and about 2500 for III-A and V-A. The question arises as to how this can occur. Since no storage of heat takes place during steady state conditions, the thermal resistance of I-A must be greater than that of the other two.

Referring now to Table A-1, information is given from which the total resistance of each configuration can be computed. For II-A, IV-A, and VI-A, for example, the total resistance is found to be 0.036 for all 3, which should be expected. For III-A and V-A, I get 0.163, which corresponds to a U value of 0.072 and is typical of construction with 3½ in. of insulation. However, in the case of I-A, the total resistance seems to be 0.136, or only 83 per cent of that for III and V. Should not all odd-numbered configurations have the same thermal resistance? This apparent inconsistency is even more perplexing when considering Fig. 11 again, because I-A, which has the smallest thermal resistance, has the lowest steady state heat flow.

A second point of discussion has to do with Fig. 5, in which the inside film conductance is the only variable. From Table A-1 and the 85 sq ft of wall area, the resistance per unit area is 10.6 for Case A. Since the reciprocal of the film conductance is the resistance, 1/1.65 yields approximately 0.6. The thermal resistance exclusive of the inside film is thus 10.0. The resistance corresponding to the conductances of 2.30 and 0.83 are 0.43 and 1.20, respectively. Hence, the total resistances for Cases D, A and E are 10.4, 10.6 and 11.8, respectively. Referred to the 1.65 coefficient, the thermal resistance of the wall has been reduced by 2 per cent in Case D and increased by 11 per cent in Case E. Evaluation of the net positive areas under the curves shows that the area under Curve D exceeds that under A by far more than 2 per cent; moreover, the heat removed in Case E is only a small fraction of that of Case A; indeed, the net heat removed for Case E is very close to zero. Since the steady state values are so markedly different for the 3 cases, some other factor or variable is necessarily responsible for the widely differing curves.

My third point has to do with the two-dimensional characteristic of the heat flow in configurations I, II and VII. The equivalent electrical circuits in Fig. 1 show that heat

cannot flow from the studs to the space-between-studs. Note, however, the results in Figs. 6 and 7. Here it is seen that the inside surface temperature over the studs is markedly different from that over the stud space. For example, in the bottom figure of Fig. 7 at 1800 hours, the inside temperature over the stud space section (t_i) is about 87 F. However, the temperature over the stud path (t_i') is only about 72.5 deg. Using the given inside surface conductance of 0.83, the heat flow per unit area is approximately 12 Btu per hr in the first case and practically zero for the second. Since plywood is a good conductor, heat should certainly flow across the interior plywood and equalize to some extent the temperatures. It appears to me that a better basis of comparison than configuration I would be the results of a two-dimensional heat flow case. This could be accomplished by connecting the lumping points in the same plane of the wall with appropriate resistances, and solving the circuit. The two-dimensional effect evidently has considerable influence, and the author is urged to pursue it further.

A small, but important, point arises as to terminology. The several figures are labeled *heat flux*, but the text includes both this and *cooling load*. The latter is understood in the industry to mean the rate at which heat must be absorbed by the refrigeration equipment. Hence, only the convective portion of the inside film coefficient contributes heat to the cooling load. The radiation portion merely transfers heat to another surface where it is absorbed and appears as cooling load at a later time. In the case of the 0.83 inside coefficient there is little or no cooling load at all, because 0.83 is the minimum radiant coefficient normally expected in a room having surfaces of 0.90 emissivity. The convective portion is therefore zero, or nearly so. Actual cooling load curves would add considerably to the value of the paper.

I want to compliment the author for attacking the many variables in such a systematic fashion by electrical analogue. We have expended a good deal of effort along the same lines except that computations were made on standard computing machines using the analytical technique developed by Dr. Nottage and George Parmelee. The mass of detail involved is very great, even without the additional complications of setting up and operating an electrical analogue. During the past summer we collected elaborate test data from an existing office building. In such structures, the large number of variables to consider imposes severe limitations on analytical techniques, and such analogues as this appear to be better fitted for the job. Because of the eventual value to the profession in being able to understand the thermal performance characteristics of structures, the author is urged to continue and expand his efforts on this important subject.

AUTHOR'S CLOSURE: It is indeed gratifying for a research worker to know that others are interested in the work he is doing, particularly when people from the industry demonstrate that interest. I should like very much at this time to thank all of the discussers who have taken their valuable time to scrutinize this paper, even though some of the discussion may be controversial.

I shall first take up Herb Nottages' comments with regard to a generalized lumping rule. A sensible method of estimating the number of lumps to be used in a network is to determine the wave-length of the temperature wave being imposed on a wall as a result of the periodic sol-air input. For sinusoidal inputs to a slab the wave length is equal to 2 times the square root of π, times the diffusivity of the material, times the fundamental period of the temperature wave, divided by the number of significant harmonics. The length of a lump should be made small compared to this wave-length. The number of significant harmonics depends upon the orientation of the wall with respect to a north-south line, the nature of the material of construction, and the precision with which answers are sought. It has been our experience that for the determination of cooling loads, considering the entire system network, reasonably good results may usually be obtained on the basis of the fundamental input only.

Both Mr. Nelson (in his Fig. A) and Mr. Gilman (in his point 3) suggest a 2-dimensional network representation accounting for heat flux perpendicular to the long dimension and laterally along the wall. As is the case for most engineering problems, the

achievement of more and more accurate predictions must be weighed against the cost of analysis and resulting savings. It is my opinion that at this point it is more important to improve our estimates of the boundary inputs, the boundary resistances and the thermal properties of the materials involved. A refinement in the thermal network to take into account vertical temperature gradients in the space might be worthwhile under some conditions. It appears that fairly simple networks representing the conduction paths allow the achievement of reasonably good engineering answers. In

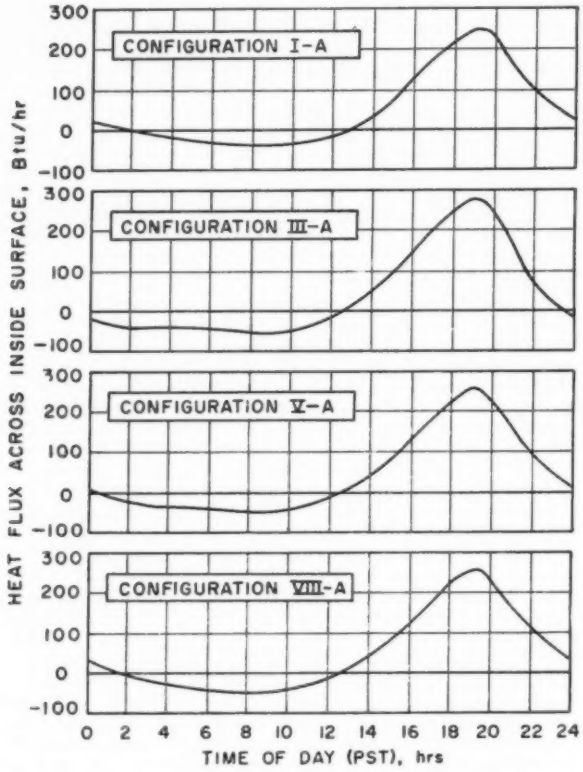

FIG. B—SAME AS FIG. 11 OF ORIGINAL PAPER WITH GRAPH FOR CONFIGURATION VIII-A ADDED

previous correspondence with Mr. Nelson it was mentioned that his network representation (in which the outside wall surfaces are considered to be isothermal with a parallel flow path through the studs and space) appears to be adequate. Shortly before leaving for Cincinnati solutions were obtained for this network configuration (VIII-A), with boundary conditions similar to the previous work. The heat flux across the inside surface plotted against time is shown in Fig. B and labeled configuration VIII-A. It can be seen that this solution compares very well with configuration I-A.

Mr. Gilman, in his discussion, noted that several points in the paper caused him concern and no wonder, it also caused me concern. Very early in our analogue work we realized that the calculation of accurate heat flux data from the temperature or potential measurements required better resolution and a differential amplifier to measure directly the difference between the inside surface and air potentials. To rectify this situation measuring equipment was obtained which would allow the grounding of the constant inside air potential resulting in a plus and minus swing of the surface potentials. The inside surface potential was then measured with respect to ground, the air potential reference, and expanded to the full width of the recorder chart.

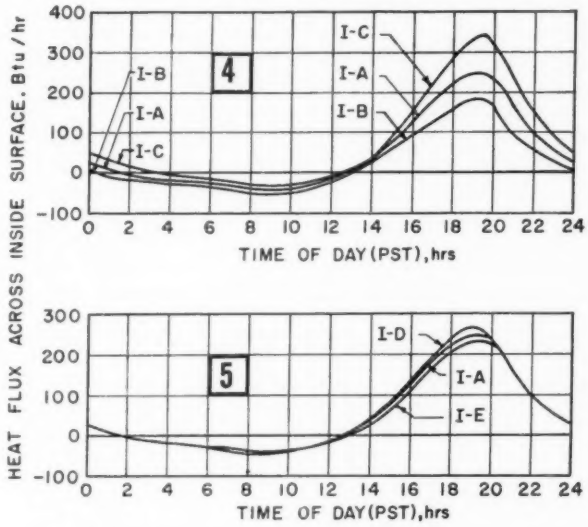

FIG. C—RECALCULATION OF DATA FROM ORIGINAL PAPER WITH 4 CORRESPONDING TO FIG. 4 OF ORIGINAL PAPER AND 5 CORRESPONDING TO FIG. 5 OF ORIGINAL PAPER

Some of the data presented in this paper have been recomputed and are presented in Figs. B and C. Fig. B corresponds to the original Fig. 11 and Fig. C corresponds to original Figs. 4 and 5. It may be noted that the net areas under the heat flux curves in Fig. B differ only slightly as would be expected.

I would like to correct one misunderstanding. Configuration I and II do not have the same value of effective steady state resistance as their simplified counterparts III and V, and IV and VI for the reason that the path through the studs was replaced by equivalent insulated or air space in the simplified networks. This is a very good assumption for the insulated wall, but not as good for the uninsulated wall.

In this paper the combined convection and radiation resistance should be interpreted as an equivalent surface to space resistance. This results in a heat flux across the wall surface as a function of time. To determine the true time-variable equipment load for a space one must consider the whole network including radiation coupling between

the conduction paths and inside heat sources and sinks. This was done in a previous paper presented at the last semi-annual meeting. More of this type of data are in preparation.

In regard to Prof. Gay's discussion, our calculations indicate the west wall sol-air input to reach a maximum at approximately 5:00 pm. The peak inside heat flux for configuration I occurs at approximately 7:12 pm resulting in a time lag of 2.2 hr for the insulated wall. For the uninsulated walls, a time lag of approximately 1 hour was obtained. From the new curves given in the lower part of Fig. C, the peak heat flux computed for configurations I-A and I-B is 250 Btu per hr and 185 Btu per hr respectively, compared to 194 Btu per hr computed by Prof. Gay. It was mentioned that other investigators generally consider GUIDE predictions to be on the high side. It is difficult to interpret this conclusion without knowing precisely under what conditions the calculations were made. I believe it is generally accepted that GUIDE predictions of cooling loads are usually high mainly because capacitive effects due to interior partitions and furnishings are not taken into account. There doesn't seem to be any good reason to expect a significant difference in prediction of heat flux through single walls using either data based on the Mackey-Wright equations or analogue computer techniques, providing the same boundary conditions, heat transfer conductances, and properties are used in both calculations.

No. 1566

ELECTRICAL ANALOGGER ANALYSIS OF A COOLED-STRUCTURE COMPLEX

By CARL F. KAYAN*, NEW YORK, N. Y.

AIR-CONDITIONED structures, or buildings maintained below atmospheric temperatures, represent a composite arrangement in which the inward flow of heat through the structure along with heat from other sources must ultimately be absorbed by some cooling medium within the conditioned space.

Absorption of the thermal loads is generally brought about through a conventional refrigerating plant wherein the heat flow to the structure is accounted for in the condenser, along with the refrigerating work of energy transport in the compressor or heat pump machine.

The present analysis is devoted to the development of a method for predicting the overall performance characteristics of a cooled structure with an integrated refrigeration plant including its attendant heat-transfer equipment. This analysis covers the total process-complex applying electrical analogger methods to the entire combined thermal circuit.

The performance prediction of a separate composite vapor-compression plant is difficult because of the many simultaneous variables acting. The design may be based on initially set conditions, but during operation the conditions may be quite different. A plant with a fixed compressor size, designed to chill a fluid at a specified initial temperature and flow rate, may operate with fluid at a different temperature. The same might be said of the heat rejection process in the condenser. If the temperature of the fluid (such as air to be cooled by the refrigerating plant) is considered the result of energy flow within a thermal structure, then the problem is further magnified. Under these circumstances, prediction of the performance of the entire cooled-structure complex, as involving the plant components of fixed dimensions, may be quite difficult. Particularly is this so if transient heat transfer enters into the problem, and if performance *vs.* time is desired.

Performance under varying operating conditions may be investigated by means of the resistance concept utilized in heat-transfer studies. Specifically, in addition, the interdependent relationships for the several system components may be investigated by electrical analogy methods with thermal values represented by electrical

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quantities. Application of the resistance concept and use of electrical analogy methods for the study of different steady-state thermal-circuit systems is well recognized[1,2,3,4,5], and likewise so for unsteady-state systems[6]. Extension of the resistance concept to include refrigerating machine performance as coupled with heat-transfer equipment has been reported on,[7,8] and although inherently a part of the present cooled-structure-complex analysis, will not be repeated here in detail.

REFRIGERATING PLANT PERFORMANCE

The basic refrigerating-plant components are: (1) vapor compressor, (2) refrigerant condenser, (3) expansion valve (pressure-drop device), (4) refrigerant evapora-

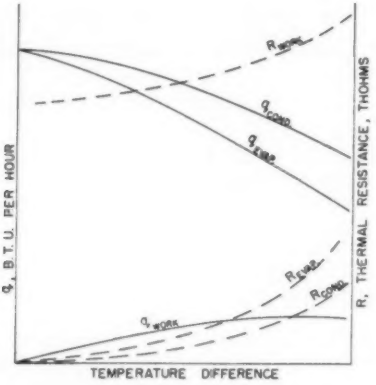

FIG. 1—TYPICAL VARIATIONS OF ENERGY FLOW RATES AND CORRESPONDING RESISTANCES AS A FUNCTION OF TEMPERATURE DIFFERENCE, FOR A VAPOR COMPRESSION REFRIGERATING PLANT

tor (direct-expansion cooling coils in the present analysis). Each component has operating characteristics which depend not only on its design, but also on the working conditions. Of prime importance in the performance of a reciprocating compressor, operating with a given refrigerant and a given rate, is the real volumetric efficiency. This, along with the low pressure conditions, governs the amount of realized refrigerating capacity. The volumetric efficiency[7,8] depends on a number of items, including clearance, nature of the gas, pressure-difference ratio, and valve pressure drops.

Typical variations of energy rates in the *heat-pump* operation of a refrigerating plant are shown in Fig. 1, as a function of the equivalent temperature difference represented by the overall difference in pressure between the evaporator and the condenser.

[1] Exponent numerals refer to References.

Ultimately, all the energy delivered at the condensing pressure and temperature must be given up in the condenser. Then the necessary condensing temperature with respect to the coolant conditions of inlet temperature and flow rate, coupled with the overall heat-transfer *coefficient* and surface area, will provide for the absorption of this energy[7,8].

The energy absorbed at the low-temperature level, under the refrigerant evaporating conditions, must in effect be transported to the condensing level of temperature. Just as the resistance concept has been extended to cover heat-exchanger energy flow via the overall temperature difference[5], so has it been further extended recently to cover energy flow in a machine[7,8]. Equivalent resistance values are used to represent compressor performance between equivalent temperature (pressure) levels. With the variation of compressor performance expressible in terms of overall temperature difference, the resistance values for the compressor energy may be obtained by dividing the temperature difference by the hourly heat quantities:

$$R = \Delta t/q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

R, the resistance value, may refer to the hourly heat for the evaporator (*i.e.* direct-expansion cooler), thus R_{evap}; hourly condenser heat, R_{cond}; or hourly compressor-work energy, R_{work}.

Typical resistance values corresponding to the energy values just indicated are shown in Fig. 1.

HEAT LOAD OF THE COOLED STRUCTURE

As noted, the cooled-structure load may consist of: (1) heat transfer through the walls[1,2,3] as based on the temperature difference between outside and inside, inclusive of surface effects, and to which may be coupled ventilation or infiltration air-cooling effect; (2) internal heat release from equipment such as lights and power; and (3) heat given off by *hot* bodies brought in from outside temperature and thus released transiently as the bodies are cooled to space temperature.

The wall heat transfer, plus the air (infiltration) cooling effect when considered together on a total hourly basis, may be represented through the structure temperature difference Δt_w by means of an equivalent resistance R_w:

$$q_w = \Delta t_w/R_w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Thus, the amount of heat flowing in, under steady-state conditions, is dependent on the temperature difference, and by the nature of the structure boundary walls and their surface effects, their total area, and by the nature and amount of the incoming air.

Internally released heat is usually independent of the maintained temperature difference, and hence is accounted for separately in the total of heat acquired within the conditioned enclosure, which has ultimately to be dissipated to the refrigerating plant. This is separately handled in the subsequent electrical analogger approach.

When a hot solid body is suddenly exposed to a cool fluid (such as air) at its surface, the heat transfer effects are transient and progressive. The temperature at the surface and at different points within the body change with time at a rate dependent on heat-transfer properties of the body and of the fluid at the surface.

The following expression, from Schack[9], satisfactorily represents the relationship for the surface temperature t_s of a hot body, originally at a uniform hot temperature

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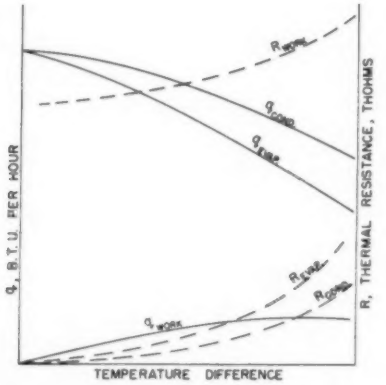

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Thus, the amount of heat flowing in, under steady-state conditions, is dependent on the temperature difference, and by the nature of the structure boundary walls and their surface effects, their total area, and by the nature and amount of the incoming air.

Internally released heat is usually independent of the maintained temperature difference, and hence is accounted for separately in the total of heat acquired within the conditioned enclosure, which has ultimately to be dissipated to the refrigerating plant. This is separately handled in the subsequent electrical analogger approach.

When a hot solid body is suddenly exposed to a cool fluid (such as air) at its surface, the heat transfer effects are transient and progressive. The temperature at the surface and at different points within the body change with time at a rate dependent on heat-transfer properties of the body and of the fluid at the surface.

The following expression, from Schack[9], satisfactorily represents the relationship for the surface temperature t_s of a hot body, originally at a uniform hot temperature

t_{ex}, suddenly exposed continuously to a cold fluid at temperature t_a and for which a surface coefficient h_a may be applied:

$$t_s = t_a + (t_{ex} - t_a)\,(f_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where f_3 is a specially defined function value depending on the material properties, the time elapsed after the initiation of exposure, and the surface coefficient (boundary conductance). Thus

$$(t_s - t_a) = (t_{ex} - t_a)\,(f_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The rate at which the surface will give up heat, based on its temperature difference *vs.* the surrounding fluid, will be:

$$q = Ah_a\,(t_s - t_a) = Ah_a\,(t_{ex} - t_a)\,(f_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

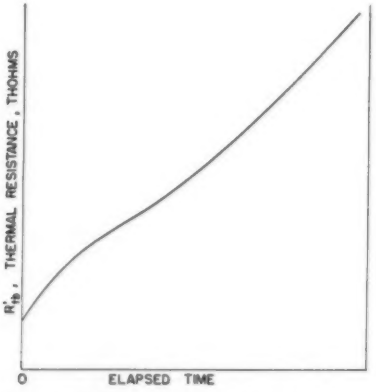

FIG. 2—TYPICAL VARIATION OF TRANSIENT BODY RESISTANCE AS A FUNCTION OF TIME

$$q/A = h_a\,(f_3)\,(t_{ex} - t_a) = (t_{ex} - t_a)/(R'_{tb}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

Hence the transient-body resistance of the material, R'_{tb} (per square foot) or R_{tb} (for the gross exposed area $= A$), will depend on time and physical properties, and be related to the original temperature difference:

$$R'_{tb} = 1/(h_a f_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$R_{tb} = 1/(Ah_a f_3) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

The typical variation for the transient-body resistance (per square foot), R'_{tb}, in accordance with Equation 6†, is shown in Fig. 2.

† It must be pointed out that the validity of such a resistance calculation, and in fact, of any transient heat-flow calculation involving an enveloping fluid around a solid body depends entirely on the assumption of the heat-transfer properties entering into the calculation. The validity of these assumptions, particularly that of the surface conductance, is always a question, making the accuracy of a refined result somewhat questionable.

HEAT EXCHANGER RELATIONSHIPS

Both the condenser and the evaporator (direct-expansion cooler) involve, in the ideal, constant temperature conditions with respect to the refrigerant (condensing or evaporating). Sensible heat with changing temperatures prevails with respect to the corresponding circulating fluids, *i.e.*, condenser coolant or evaporator fluid. Typical heat exchange curves for these equipments are shown, combined in Fig. 3.

The flow of heat in the condenser may be considered to vary with the overall temperature difference between the saturation condensing temperature and the incoming coolant temperature, with the effect of compression superheat on condenser performance neglected. The hourly condenser-heat flow, as given in a previous paper[5], may be shown in terms of resistance:

$$q_c = \Delta t_{t\text{-}c}/R_{t\text{-}c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

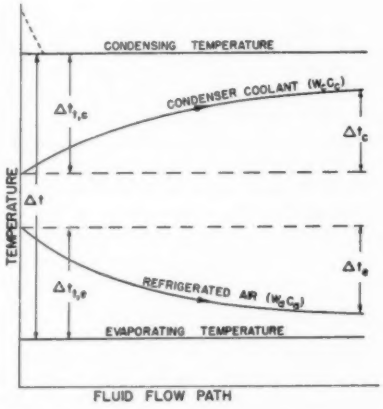

FIG. 3—TYPICAL TEMPERATURE CURVES FOR REFRIGERATING PLANT HEAT TRANSFER EQUIPMENT

$$R_{t\text{-}c} = \frac{R_c}{1 - \dfrac{1}{e^{\frac{R_c}{R_{UA\text{-}c}}}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

$$R_c = 1/(W_c C_c) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$\Delta R_{t\text{-}c} = R_{t\text{-}c} - R_c \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

These relationships are depicted on a general basis in Fig. 4, plotted against coolant flow rate, W_c.

Similar equations apply to the direct-expansion cooler (evaporator), for the general temperature relationships shown in Fig. 3:

$$q_e = \Delta t_{t\text{-}e}/R_{t\text{-}e} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

$$R_{t\text{-}e} = \frac{R_a}{1 - \dfrac{1}{e^{\frac{R_a}{R_{UA,e}}}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

$$R_a = 1/(W_a c_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

$$\Delta R_{t\text{-}e} = R_{t\text{-}e} - R_a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Since these equations are similar to those of the condenser, Fig. 4 may be considered to typically represent them also.

STRUCTURE-COMPLEX PERFORMANCE CHARACTERISTICS VIA ELECTRICAL ANALOGGER

The thermal energies acquired in the structure must ultimately be taken up by the refrigerant, and dissipated, along with machine work, in the condenser. These structure energies flow into the system via the evaporator (direct-expansion cooler). They thus enter the system as a given fluid weight-flow-rate cooling through the requisite temperature difference. They may be considered to enter the cooling system at the temperature that corresponds to the space-system equilibrium conditions. This means equilibrium between entering-structure energy and evaporator-extracted energy. As just shown, the different energy flow rates may be defined in terms of resistances and resultant temperature differences.

The different energy resistances are interrelated in the overall process complex. They may thus be represented together in a thermal-energy circuit of energy flow in the structure, in the evaporator, and in the condenser, as well as the energy transport in the compressor.

For practical manipulations, translation from thermal to electrical circuit values proves advantageous.

The electrical analogger circuit may be constructed to facilitate the performance evaluation of the overall structure-complex, including the refrigerating plant. Herein electrical resistance values, proportional to those set by the preceding thermal equations, are employed, with electrical voltage‡ the counterpart of steady-state temperature difference.

In the present type of analysis, many simplifying assumptions are expedient. Idealized performance for the sake of clarification is assumed, although corrections to account for the performance of real equipment could be made. The essential purpose here is to develop further tools for use in electrical analogger operations.

Whatever energy is discharged from the compression machine as transport energy, it is important to note, must in turn flow through the condenser system. It must further be emphasized that this compression-machine energy is made up of the heat flow to the evaporator, plus the work-energy of compression. The extent of the temperature spread above the entering coolant temperature, as required by the resultant condenser-heat flow, will in turn determine the condensing-vapor temperature, and thus, in effect, the compressor delivery pressure.

Similarly, in terms of the structure air-circulation conditions of temperature and flow rate, the requisite temperature difference necessary for the evaporator heat transfer must establish the evaporator temperature. In this way the corresponding evaporator pressure—the suction pressure of the compressor plant—is established. The actual compressor energy, depending on real volumetric efficiency and hourly

displacement, is subject to the terminal pressures of the system. It is subject, thus, to the condensing pressure (as determined by the condenser heat transfer), and to the evaporator pressure (as determined by the evaporator heat transfer).

The structure heat flow must be absorbed by the evaporator. In self-regulating the internal structure temperature under off-design conditions, the control of the amount of heat which must be picked up by the evaporator is effected. Thus, there will result some internal structure-temperature, wherein the load can be picked up by the evaporator, pumped to the higher temperature level of the condenser, such that it may appropriately be rejected to the condenser coolant.

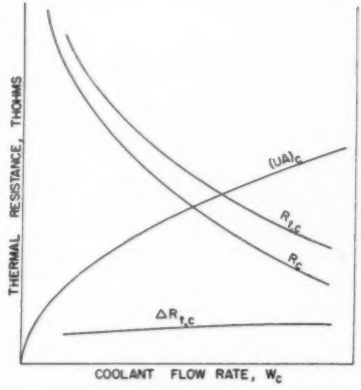

FIG. 4—TYPICAL VARIATION OF CONDENSER RESISTANCE, AND "UA" VALUES, AS A FUNCTION OF FLOW RATE

It is then apparent that the structure-refrigerating plant complex, with all its components and their operating characteristics, presents a rather complicated thermal-circuit problem with respect to prediction of off-design performance.

ANALOGGER CIRCUIT

Fig. 5 shows the complete electrical analogger circuit for the process-complex, combining the structure thermal circuit as an integral part of the refrigerating-plant thermal circuit.

The analogger circuit is set up as a network involving several parallel branches. It comprises 2 separate and adjustable resistances, in parallel, to represent the work- and the evaporator-heat phases of the compression machine system. The total current contributed by these 2 branches must equal that flowing through the condenser circuit, with the top potential common to both condenser and machine sections. The potential value corresponding to the coolant-inlet temperature at point 4 is

‡ Use might be made of a variable-voltage electrical generator to represent the compressor-machine characteristics with respect to varying effective temperature difference. It seems simpler, however, to use an impressed overall voltage; as on the paralleling condenser system.

maintained on the condenser circuit by means of an adjustable ballast resistance $r_{v,4}$. The evaporator heat-transfer section, the counterpart of the condenser heat transfer section, is separately established as an integral part of the structure-cooling circuit. In operation, its energy flow must be matched by that through the evaporator branch of the machine circuit, as attested by the corresponding current meters, I_e.

Both the condenser and the evaporator heat-transfer total resistances, $r_{t,c}$ and $r_{t,e}$, include their respective fluid-exchange resistances r_c and r_e. Particularly to be

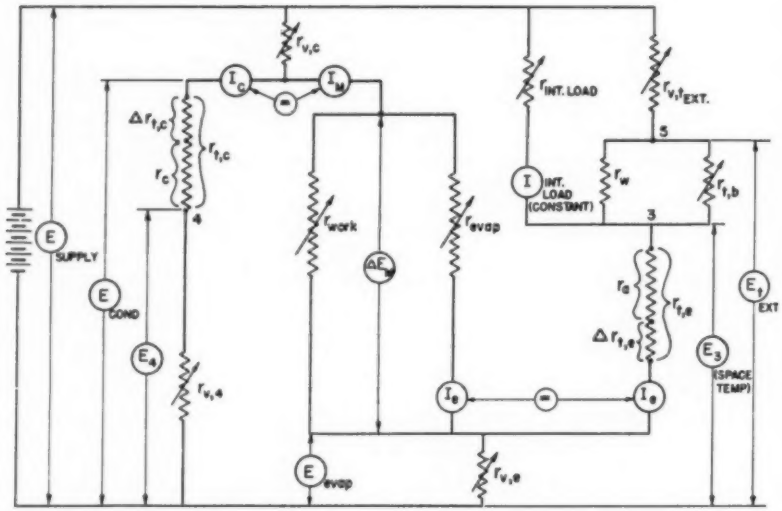

FIG. 5—ELECTRICAL ANALOGGER CIRCUIT FOR THE PROCESS-COMPLEX OF REFRIGERATING PLANT AND COOLED STRUCTURE

noted is that the external-air-temperature potential at point 5 is maintained at some specified value as required by t_{ex}. The value of the transient-body resistance r_{tb} is set as corresponding to the particular point of time for which the study is made. Control to bring the evaporator current meters I_e into agreement is made by means of an adjustable *evaporator* ballast resistance $r_{v,e}$. This governs the bottom evaporator potential, with aid through trimming corrections on the machine variable resistances. The resultant potential difference between the evaporator and the condensing-vapor potential, representative of the machine pressure span, is in turn utilized to determine the effective resistances of the compressor elements r_{evap} and r_{work}. The base or ground potential of the system is taken as 0 F.

Adjustment of the analogger circuit is made progressively through change in the condensing-vapor potential by the variable resistance $r_{v,c}$. This must establish the condenser current I_c equal to the compressor current I_m, while simultaneously keeping the potential at point 4 (coolant-inlet temperature) at its proper value, along

with the potential at point 5 (external air temperature) at its specified value. Coincidental values of r_{evap} and r_{work} are maintained at their particular values as dictated by the machine potential difference ΔE_m. As noted previously, the evaporator current meters are kept alike in value.

The current meters show the equivalent energy flowing at given points; the potentials show the equivalent temperatures. Adjustment of this type of integrated

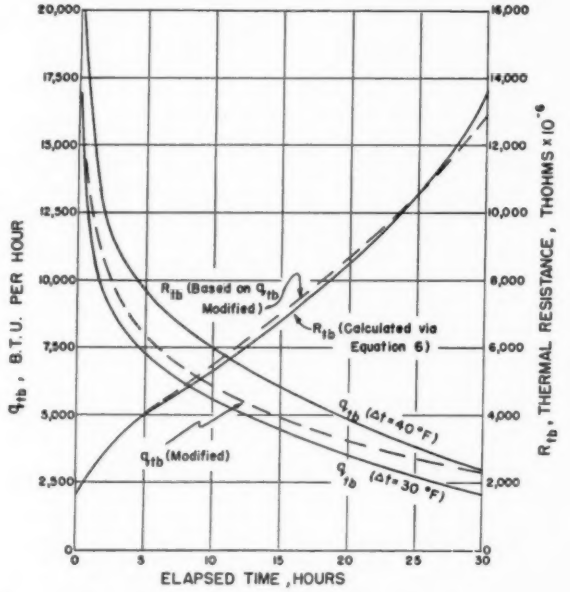

FIG. 6—TRANSIENT BODY HEAT FLOW AND EQUIVALENT RESISTANCE AS A FUNCTION OF TIME

heat-power analogue, it must be recognized, is a matter of manual dexterity, somewhat simplified by the use of automatic equipment, a voltage regulator for example.

ILLUSTRATIVE EXAMPLE

An insulated structure, 40 × 20 × 10 ft, designed for industrial air-conditioned process purposes, is to be maintained at 55 F under conditions of 95 F outside, with water supply for the condenser at 70 F. The walls, of light-weight construction with reflective insulation, have negligible heat capacity. Including an allowance for incoming ventilation air, the total structure heat flow may be taken as 700 Btu per (hr) (F deg). An internal heat load due to power and lights is specified as 15,500 Btu per hr. The influence of the introduction of a hot load initially at a uniform temperature of 95 F, on the structure-space temperature is to be investigated as a function of time.

The hot load may be considered as a tightly packed stack 20 × 6 ft, and 2 ft thick, having the following uniform properties: density ρ, 94 lb per cu ft; specific heat c_b, 0.22 Btu per (lb) (F deg); and thermal conductivity k, 0.70 Btu per (sq ft) (hr) (F deg per ft); together representing a thermal diffusitity a of 0.034. The surface conductance h_a is assumed to have the general value of 2 Btu per (sq ft) (hr) (F deg).

For the refrigeration plant: compressor displacement, 11.2 cfm; condenser heat-transfer surface, 16.85 sq ft; condenser coolant-inlet temperature is 70 F, and its

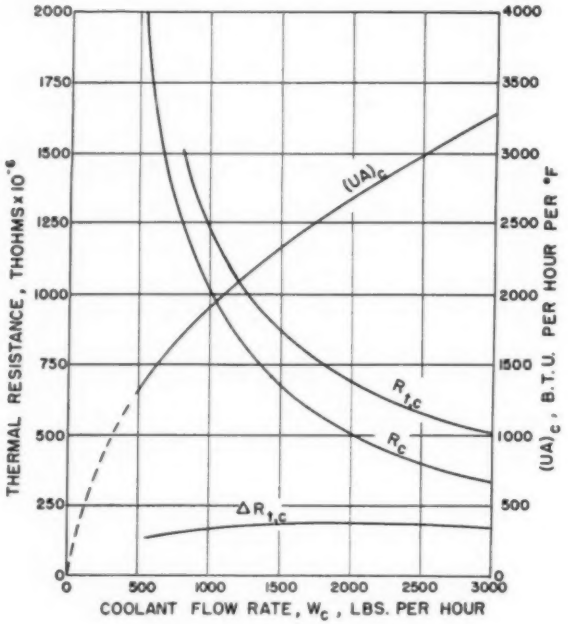

FIG. 7—CONDENSER PERFORMANCE CURVES AS A FUNCTION OF COOLANT FLOW RATE

coolant flow rate W_c at 1800 lb per hr. The structure-space air circulation rate over the air-cooling coils (evaporator) is to be taken at 7000 lb per hr, corresponding to about 1600 cfm, and specific heat, $c_a = 0.24$.

In accordance with the provisions of Equation 5, and the specified heat-transfer properties for the *hot* load material, Fig. 6 has been developed. This shows the variation of heat dissipated off the surface for 2 air temperatures, namely 55 and 65 F, as a function of time. Since at the beginning of exposure to the lower room-temperature-air the heat loss is rather high, tending quickly to raise the structure-air temperature, and also in view of the general indefiniteness of the value for h_a in practice, it is regarded as desirable to use a modified curve for the heat loss, as in-

dicated in Fig. 6. The corresponding value for the transient-body gross resistance R_{tb} is likewise shown. The temperature difference for the transient-body resistance R_{tb} may be assumed as for the wall*, *i.e.* Δt_w.

Resistance values for the condenser, to correspond with those of Equations 7, 8, 9, and 10, are directly obtained from Fig. 7, typified originally by Fig. 4. In Fig. 7, values for the product UA_c are included, all values being plotted against W_c, the hourly rate of condenser water flow.

Corresponding to the air flow rate of 7000 lb per hr, and in terms of Equation 13, $R_a = 595 \times 10^{-6}$ thohms; via Equation 12, $R_{t,e} = 752 \times 10^{-6}$ thohms; and via

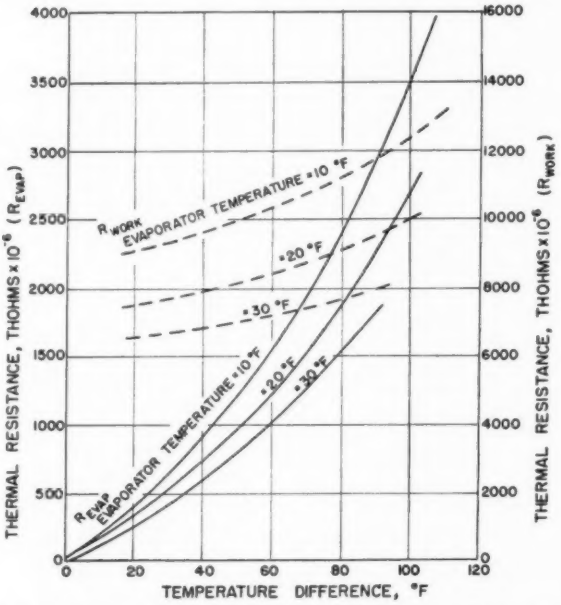

FIG. 8—COMPRESSOR PERFORMANCE CURVES AS A FUNCTION OF GENERATED TEMPERATURE DIFFERENCE

Equation 14, $\Delta t_{t,e} = 157 \times 10^{-6}$ thohms. These air values are the counterpart of those for the condenser in Fig. 7.

Hourly heat values, as typified by Fig. 1, taking real volumetric efficiency into consideration, are utilized to develop Fig. 8 for machine resistance values, as plotted against equivalent temperature differences between evaporating and condensing conditions. Equivalent resistance values, R_{evap} and R_{work}, for the compressor-machine performance in accordance with Equation 1, are specifically shown in Fig. 8, as thohms.

* Whereas conventionally in transient heat flow analysis, account must be taken of heat storage effects—represented by electrical capacitance in usual electrical analogy studies—here the transient process may be adequately handled through the resistance concept, using effective resistance values changing with time.

The equivalent thermal resistance for the wall plus ventilation air, and neglecting wall heat-storage effects, is to be taken at $R_w = 1430 \times 10^{-6}$ thohms, and constant. The constant internal heat load is specified at 15,500 Btu per hr.

Electrical circuit values, as translated from Figs. 6, 7, and 8, the wall and internal heat values, and the specified-evaporator-air-cooling data according to Equations 12, 13, and 14 specified for use in the analogger arrangement of Fig. 5, are on the basis of 1 ohm electrical resistance = 10^{-6} thermal resistance thohms, 1 volt = 10 F, and 1 milliampere = 1000 Btu per hr.

Fig. 9 shows the variation of the structure-space temperature as a function of time, in hours, after the introduction of the *hot* bulk load into the cooled space, determined stepwise for different time intervals.

ACKNOWLEDGMENT

Acknowledgment is gratefully made to associates of the author, particularly Prof. J. A. Balmford of the Electrical Engineering Department, Columbia University.

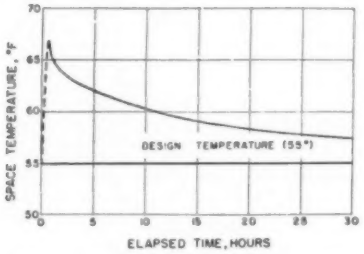

FIG. 9—VARIATION OF COOLED-STRUCTURE SPACE TEMPERATURE AS A FUNCTION OF TIME

This paper reports the results of another phase of the investigations supported in part by a grant-in-aid to Columbia University from *The Refrigeration Research Foundation*.

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NOMENCLATURE

A = heat transfer surface area, square feet.

c_a = evaporator air flow specific heat, Btu per (pound) (Fahrenheit degree).

c_b = transient body specific heat, Btu per (pound) (Fahrenheit degree).

c_c = condenser coolant mean specific heat, Btu per (pound) (Fahrenheit degree).

a = thermal diffusivity, $= k/(\rho c_b)$.

f_3 = function of $4a\theta/L^2$ and $h_a L/(2k)$.

h_a = boundary conductance of air (heat transfer), Btu per (sq ft) (Fahrenheit degree) (hour).

L = thickness of transient solid body, feet.

k = transient body thermal conductivity, Btu per (square foot) (hr) (Fahrenheit degree per foot).

ρ = density, pounds per cubic foot.

θ = time elapsed, hours.

q = heat energy as applied to different elements of system, Btu per hour and identified by subscripts, such as in conjunction with compressor, *i.e. work*, *evap*, and *cond.*

R_c = condenser-coolant heat-exchange resistance, thohms, Fahrenheit degree per (Btu) (hour).

R_a = evaporator-air-flow heat exchange resistance, thohms, Fahrenheit degree per (Btu) (hour).

$R_{t.c}$ = total equivalent thermal resistance for condenser, thohms, Fahrenheit degree per (Btu) (hour).

$R_{t.e}$ = total equivalent thermal resistance for evaporator (air cooler), thohms, Fahrenheit degree per (Btu) (hour).

$R_{UA.c}$ = net condenser heat-transfer resistance, thohms, $= 1/UA_c$.

$R_{UA.e}$ = net evaporator heat-transfer resistance, thohms, $= 1/UA_e$, Fahrenheit degree per (Btu) (hour).

r = electrical resistance corresponding to thermal resistance, ohms.

Δt = temperature difference, Fahrenheit degrees, between evaporator and condensing temperature.

$\Delta t_{t.c}$ = temperature difference, Fahrenheit degrees, between condensing vapor and coolant inlet temperature, also as indicated by special subscript.

$\Delta t_{t.e}$ = temperature difference, Fahrenheit degrees, between evaporating vapor and air-flow inlet temperature.

U = overall heat-transfer conductance, (coefficient), Btu per (square foot) (hour) (Fahrenheit degree), applicable, with subscript to either condenser or evaporator.

W_c = condenser-coolant flow rate, pounds per hour.

W_a = evaporator-air flow rate, pounds per hour.

DISCUSSION

H. T. GILKEY, Cleveland Ohio (WRITTEN): This paper outlines another of the many uses of the analogue analysis of heating and cooling loads. It includes an approach which can be used without the necessity for input wave generators. This should broaden the use of the analogy, even though, as the author points out, the ease of operation is dependent upon the manual dexterity of the operator.

The primary disadvantage of the resistance analogue may be that the current flow must be in one direction in all sections of the circuit. This means that heat storage effects can be taken into account only during periods of uni-directional heat flow, and transient, bi-directional heat flow resulting from this storage cannot be analyzed. Furthermore, this characteristic of the analogue may reduce its usefulness in analyzing the

effects of moisture storage and release in hygroscopic building materials and furnishings. Of course, adjustable-voltage power supplies could be substituted for appropriate resistors under these conditions, but this would reduce the simplicity of the approach.

Nevertheless, the resistance analogy should prove valuable for determining loads at maximum and near-maximum conditions. For most applications, these are the conditions in which we are most interested.

Additional work is needed on the methods of introducing the characteristics of non-homogeneous walls into analogue procedures. In many applications, transmission loads through building materials are relatively unimportant because of the magnitudes of internal loads. In structures having non-homogeneous walls, however, the opposite is often true. This is the case with residences, for instance, in which the internal loads due to occupancy, lighting and cooking for example, are usually small compared to the transmission loads. Thus, information on using the analogue is not complete for the application in which it could well prove most useful.

S. F. GILMAN, Syracuse, N. Y. (WRITTEN): This paper adds another technique to those at hand for analyzing thermal circuit problems. It seems strange to think of a compressor being represented by a thermal resistance, but apparently it can be handled quite satisfactorily in this manner.

The analogue circuit is of particular interest. It is notable that no capacitances appear in the circuit, yet a case of transient heat flow is handled. This is accomplished by replacing the transient condition with a succession of steady states. Voltages are measured; then the resistances are adjusted, and voltages again measured for the new steady state condition. Such an analogue is of utmost simplicity and certainly must be inexpensive to construct. The major operating difficulty is making the proper manual adjustments to the rheostats.

The limitations of the equipment should be pointed out. The simpler the analogue, the simpler must be the problem to be solved. Consequently, when extended to complex problems it becomes necessary to make enough simplifying assumptions so that the problem can be fitted to the analogue. In the illustrative example presented, therefore, the outside temperature is taken as constant for the 30 hours. The walls of the structure are considered to have zero heat capacity, whereas in actuality walls, ceilings and floors will have considerable heat capacity. Moreover, with such a rapid change of ambient temperature as occurs in the first hour or two, considerable heat would be stored in the walls during the initial portion of the transient. Inclusion of thermal capacitance effects would evidently have considerable influence on the shape of the Fig. 9 curve.

One aspect is not entirely clear to me. Referring to Fig. 6, the heat flow from the hot body is shown for ambient temperatures of 55 F (Δt = 40 deg) and 65 F (Δt = 30 deg). Also shown is a *modified* heat flow curve from which is obtained a *modified* thermal resistance curve for the transient body. From the modified curve it is evident that the assumption is made that the ambient temperature will be 65 F after about 1 hour and 55 F after 30 hours. This implies advance knowledge of the results to be obtained from the analogue. When the resistance values from Fig. 6 are put into the device, the curve of Fig. 9 is obtained. It seems to me that we are anticipating the answer, putting it into the analogue, and getting nearly the same answer back out. Professor Kayan is asked to amplify the reasoning leading to the selection of the modified curve. There is also the question of the validity of applying transient heat flow data for constant ambient temperature conditions to the case of a variable ambient temperature.

It would have been desirable to show the refrigeration equipment performance, especially compressor work, over this 30-hr period. Just after the hot body is introduced, the compressor work is undoubtedly many times that at the end of the 30-hr period. The practical implication is that a very large compressor would be called for or, alternatively, a smaller compressor would be severely overloaded during the initial portion of the transient condition.

Because of the simplifying assumptions necessary for the example presented, the valuable next step would be to compare the analogue solution with actual experimental data

collected for a particular problem. It is analogues like this that we hope can eventually be used to provide us with application engineering data. Use of such a tool, however, must await proof of (1) that the analogue represents as closely as needed the physical configuration being studied; and (2) that the analogue yields solutions which agree with data collected under actual field conditions.

L. W. NELSON, Hopkins, Minn. (WRITTEN): It is interesting to note how rapidly the space temperature increases and the length of time required to bring the temperature back to design conditions after inserting a hot load in a cooled space. Perhaps if capacitance of the building itself were considered the initial increase in space temperature would not be as great but perhaps at the same time would lengthen the time required to obtain design conditions again.

I would be interested in knowing how the author obtained the surface conductance of 2. Is this the combined conductance of convection and radiation? I would expect the convection coefficient to be in the neighborhood of 0.75 for free convection. I would also like to ask if the author thinks it would be practical to consider the heat transfer coefficient as a function of temperature difference in his construction of the curves of Fig. 6.

Seeing that the modified curve of Fig. 6 is based upon the fact that the temperature difference between the hot load and the space temperature does not remain constant, should the modified curve be based upon a trial and error procedure? In other words, since the heat released by the hot load is a function of temperature difference and the temperature difference is a function of the amount of heat released by the hot load, the 2 are interrelated.

H. B. NOTTAGE, Encino, Calif. (WRITTEN): Under what conditions will it be impractical or impossible to employ variable resistors, and the potentials derived therefrom, as in Fig. 5, to adequately analyze the performance of a complete thermal system involving sources, sinks, resistances, and capacitances arranged as in a complete air-conditioned structure?

What is the difference in concept and operation between the author's analogger, as applied to this type of problem, and other analogue computers?

AUTHOR'S CLOSURE: I am deeply gratified and appreciative for these thoughtful and thought-provoking written discussions, because I know that their authors had to take considerable time from their busy preparatory period for coming here in order to reduce their comments to writing. May I therefore say, thank you heartily.

As so often happens in the development of further steps in the analogue treatment, we get into subsequent complications of new work programs to solve the additional problems. This is the situation here. As I mentioned right at the beginning, further work on the study of the behavior of the different components of complete thermal circuits dealing with the performance characteristics under the different conditions is in progress. We look forward to carrying on the work in conjunction with structure and internal component parts, particularly with respect to heat capacity in terms of thermal resistance, such as represented here by the present hot body brought into the space to be cooled.

We believe that we can study the characteristics of the different components to represent the performance relationships and then crank them into this computer. As noted right at the beginning with the analogger as a computer device, the results that we come out with in the end can only be measured in terms of the input validities; hence we emphasize that we have had to make some simplifying assumptions. Sometimes they are rather crass assumptions; but I would submit that if we brought a hot body, such as described in the paper, into this very room, set it down here, and left each of you to predict how that hot body would cool off and what the actual behavior of that hot-body temperature would be as a function of time, we would all come up with different answers. What we perhaps ought to do is actually take such a hot body and run a field test on it and also on other components. I assumed, as one of my licenses, that I could show the curve of variation of load as a function of time on the basis of $h = 2.0$. I really don't know whether h would be some other number such as 1.5. I wouldn't be able to tell

what the actual velocity effects would be over the surface of this hot body and what the actual emissivity effects would be. It is only a prediction; it is in terms of such that I say to Mr. Nelson in bringing out this point that I just assumed $h = 2.0$.

I have recognized that perhaps the resistance will vary with the extent of the temperature difference. I concluded, therefore, since this is a gamble and a guess, I would see what the variation would be for taking 40 deg and 30 deg, and I would take my modified curve on the basis of these 2 temperature differences.

I had a general idea that the spread in temperature might be something on that order. That is the background for the answer to Dr. Gilman, as to why I used such a modified curve. In a way, this procedure sort of takes an estimated result and cranks it back in. It also uses intuition but frankly I couldn't tell you what the answer would be on a hot body brought into a space, and I make mention of this dubious situation in a footnote in the paper. The actual behavior is only subject to a field study in terms of the thermal properties of the material and in terms of the surface conditions. This I think we must keep in mind.

In comparison with the other types of analogues as presented in the question by Dr. Nottage, there are, let us say, in general and in active use, a variety of electrical analogues for component studies, one such as was ably presented this morning by Professor Buchberg. We have also the quite famous Leopold hydraulic analogue which has been presented here a number of times. The performances as reflected on studies of such analogues would be performances for components.

The objective here was to introduce a new tool for the complete circuit, not just a portion of the circuit. It is always necessary to consider actual performances as predicted by these other analogues and they all tie in together. Those results could be employed on the resistance basis, I believe, in the present complex type of analogue.

I think I have answered Mr. Gilkey's questions on the effects of the wall structure characteristics. They were also presented in another way by Dr. Gilman's and Mr. Nelson's questions, which I believe I have answered in addition to Dr. Nottage's.

As I say, when we get into this sort of thing, we make additional future work programs for ourselves, but it is a lot of fun. Again I would like to pay my respects to the discussers for their very thoughtful and thought-provoking contributions. I hope I have satisfied the different questions.

1567

No. 1567

INDEX FOR EVALUATING HEAT STRESS IN TERMS OF RESULTING PHYSIOLOGICAL STRAINS

By H. S. BELDING* AND T. F. HATCH†, PITTSBURGH, PA.

HEAT is a stress faced by all human beings. It is therefore not surprising that physicists, engineers and climatologists have sought understanding of the physical components of heat stress, and that the consequences of exposures to heat have been examined extensively by physiologists and psychologists. For many situations, the magnitude of environmental heat load on man can now be determined, and the consequences of exposure can be predicted. However, there is yet no really satisfactory system for grading heat stress in terms of the probable strains or consequences of exposure.

To qualify as satisfactory a system would have to be simple both to employ and to comprehend, yet would have to take into consideration all determining basic factors. The difficulty lies in the large number of such factors. Thirteen are listed as follows:

Factors determining imposed stress are: (*a*) air temperature; (*b*) temperature of solid surroundings (walls); (*c*) temperature of the skin; (*d*) vapor pressure at the skin; (*e*) environmental vapor pressure; (*f*) air speed; (*g*) body heat production; (*h*) body surface area and postural attitude; and (*i*) clothing.

Factors determining resulting strains are: (*a*) capacity for circulation of blood to the skin and bodily consequences of such circulation; (*b*) capacity for maintaining a wetted skin by sweating and bodily consequence of such sweating; (*c*) tolerance for elevation of body temperature and bodily consequences of such elevation; (*d*) duration of exposure.

It is obvious that a simple system of grading heat stress in terms of resulting strain cannot treat all of these items as separate variables. The problem is, then, to find and relate the primary determining variables, and to fix the others or eliminate them from consideration.

In the light of these considerations, it seems appropriate to examine the adequacy of the best available systems of grading heat. The first is the widely used system

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of effective temperatures developed by the ASHAE. From the physiologist's point of view, it seems remarkable that the effective temperature scales, based on immediate psychological reactions as to thermal equivalence of different environments, should so well predict combinations of conditions which result in equivalent long-term physiological effects. The principal deficiencies of these scales are: (1) that higher levels of body heat production are not covered; (2) that the meaning of the units of the scales in terms of physiological strains is not apparent; and (3) the scale units do not indicate equal increments of physiological strain. This last is because they are based empirically on equal units of temperature of saturated air.

A second system of grading, the *Probable Four Hour Sweat Rate* (P4SR) of McArdle[1] et al, is less well known, but comes closer to meeting the requirements for a scale which indicates stress in physiological terms. Its main disadvantages are that it too is empirically constructed and that the significance of its units in terms of physiological strain is not obvious.

This communication is devoted to the propositions (1) that values for components of heat stress can be rationally combined in a way which will predict the physiological strain resulting from an exposure, and (2) that a simple scale constructed in accordance with this relationship will be easier to interpret and to use than the empirically designed scales that are presently available.

The aim is to provide an index of heat stress based on strains predicted to result from exposure. The approach is: (1) to provide a means for evaluating the components of heat stress of any environment; (2) to determine what, if any, systematic relationship exists between these components of stress and the resulting physiological strains; (3) to assign to the stresses index values which have meaning in terms of human endurance for heat.

EVALUATION OF COMPONENTS OF HEAT STRESS

Man is a homeotherm, *i.e.*, he maintains a constant body temperature. This implies that he has mechanisms which operate, within their capacities, to maintain a balance between heat production and dissipation. Under conditions of ordinary thermal *comfort*,

$$M + R + C \cong 0 \ddagger \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

M = bodily heat production resulting from metabolism.
R = radiation gain by the body.
C = convective gain.

Since M is always positive, $R + C$ must be negative by an equal amount. R and C are maintained in balance with M by vasomotor adjustment of blood flow to the skin.

Heat stress is considered to exist whenever, despite vasomotor adjustment, metabolic heat production exceeds the combined losses by radiation and convection; in other words, when $M + R + C$ gives a positive value.

Heat Load: It follows that the primary component of heat stress is the load which

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[1] Exponent numerals refer to References.

‡ Strictly, $M + R + C + E_{insens} \cong 0$, where E_{insens} is the 50 to 100 Btu per hour of *insensible perspiration* which is lost by evaporation of moisture put into the breath and diffused through the skin. E_{insens} is neglected in this treatment.

TABLE 1—ESTIMATES OF ENERGY METABOLISM (M) OF VARIOUS TYPES OF ACTIVITY
(Values apply for a 154-lb man, and do not include rest pauses)

		M BTU/HR
	Sleeping	250
	Sitting Quietly	400
LIGHT WORK	Sitting, moderate arm and trunk movements (*e.g.*, desk work, typing)	450–550
	Sitting, moderate arm and leg movements (*e.g.*, playing organ, driving car in traffic)	550–650
	Standing, light work at machine or bench, mostly arms	550–650
MODERATE WORK	Sitting, heavy arm and leg movements	650–800
	Standing, light work at machine or bench, some walking about	650–750
	Standing, moderate work at machine or bench, some walking about.	750–1000
	Walking about, with moderate lifting or pushing	1000–1400
HEAVY WORK	Intermittent heavy lifting, pushing or pulling (*e.g.*, pick and shovel work)	1500–2000
	Hardest sustained work	2000–2400

must be dissipated by means other than R and C if heat balance is to be maintained. The available *other means* is the secretion and evaporation of sweat. Therefore, the load may be expressed in terms of evaporation required for heat balance, E_{req}.

$$M + R + C = E_{req}$$

Determination of heat load, E_{req}, depends on adequate estimates of M, R and C. Under many circumstances, these estimates can be made. The metabolic cost, M, of many kinds of activity has been reported in studies scattered through the physiological literature,[2,3] and coefficients of heat transfer by R and C have been established for man.

M values for various representative types of activity have been assembled in Table 1.

Haines and Hatch[4] have recently provided a quick means for separately estimating R and C for an average-sized nude man with a skin temperature of 95 F, based only on knowledge of air temperature, globe temperature and air speed. The coefficients were obtained from work of Nelson et al[5] at Fort Knox. The separate values in Btu per hr may be approximately determined from

$$R = 22\ (t_w - 95)^* \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$C = 2\sqrt{V}\ (t_a - 95) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

t_w = mean temperature of the solid surroundings (walls).
t_a = ambient temperature.
V = air speed in feet per minute.

* Use of this rather than the true fourth power relationship for heat transfer between walls and man will result in an error in estimating R of not more than 50 Btu per hr, unless wall temperature exceeds 175 F. T_w is calculated from globe temperature (T_g), air temperature and air speed, as follows:

$$T_w = \sqrt[4]{T_g^4 + (0.103 \times 10^9 \sqrt{V})(T_g - T_a)}$$

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The sum of $R + C$ may also be obtained directly from the globe temperature and the air speed, as in chart A of Fig. 1. This assumes that the globe temperature represents the weighted average of the wall and air temperatures, in accordance with the following equations:

$$R + C = (22 + 2\sqrt{V})(t_g - 95) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and

$$t_g = (k_r t_w + k_c \sqrt{V} t_a)/(k_r + k_c \sqrt{V}) \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The assumption is in error in proportion to the differences in value of the heat transfer coefficients for man and globe. Fortunately, the radiation difference compensates for the convection difference and the value of $R + C$ obtained from chart A agrees within 5 percent with the sum of the separately calculated values. Total heat load, $M + R + C = E_{req}$, is found by entering chart B of Fig. 1.

When heat load is estimated by this method, four of the applicable factors determining imposed stress are used as determining variables, three are fixed.

One of the fixed items is body surface area, at 20.0 sq ft (applicable for a man about 5 ft 8 in. tall who weighs 154 lb). Actually, a small man is less stressed than a large one, and a thin man less than a fat one because of differences in heat production and surface area and the ratio between the two.

The second fixed item is skin temperature. Under different hot conditions endurable for prolonged periods, mean skin temperature has been found to lie between extremes of about 94 and 98 F. Actually, temperatures of 96 or 97 F are probably more representative of fairly severe conditions than the 95 F used by Haines and Hatch[4]. The 95 F value is used here since it is applicable for many conditions of moderate stress and since the resulting overestimate of stress under severe conditions will compensate in part for the underestimate resulting from use of this system on clothed men.

Clothing is the third variable fixed for the estimate, and it is unfortunate that limitations of available knowledge make it necessary to fix on a no-clothing basis. Since clothing acts as a thermal barrier between the skin and environment, estimates of load may be seriously in error if heavy clothing is worn. For single layers of light weight clothing, it is probable that the resulting underestimate of load below air and wall temperature of 95 F and the overestimate above will not represent serious errors.

Heat Acceptance of the Environment: The other component of heat stress is the resistance of the environment to actual evaporation of the sweat required for thermal balance, E_{req}. This component is also secondary in that it does not affect the heat load, but does affect its dissipation.

The capacity of the environment to accept the load depends on the vapor pressure gradient between the skin and air, and on the air speed. Haines and Hatch[4] have also provided a simple method for estimating maximum evaporative capacity, E_{max}, again on the basis of constants from the Fort Knox studies, and under the conditions listed above, with the added assumption that the skin is everywhere maintained wet and is at 95 F (skin vapor pressure, 42 mm Hg). Here,

$$E_{max} = 10.3\, v^{0.4}\,(42 - VP_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where

VP_a = the vapor pressure of the air in millimeters of mercury.

When dry- and wet-bulb temperatures and air speed are known, E_{max} may be determined from charts X and Y of Fig 1.

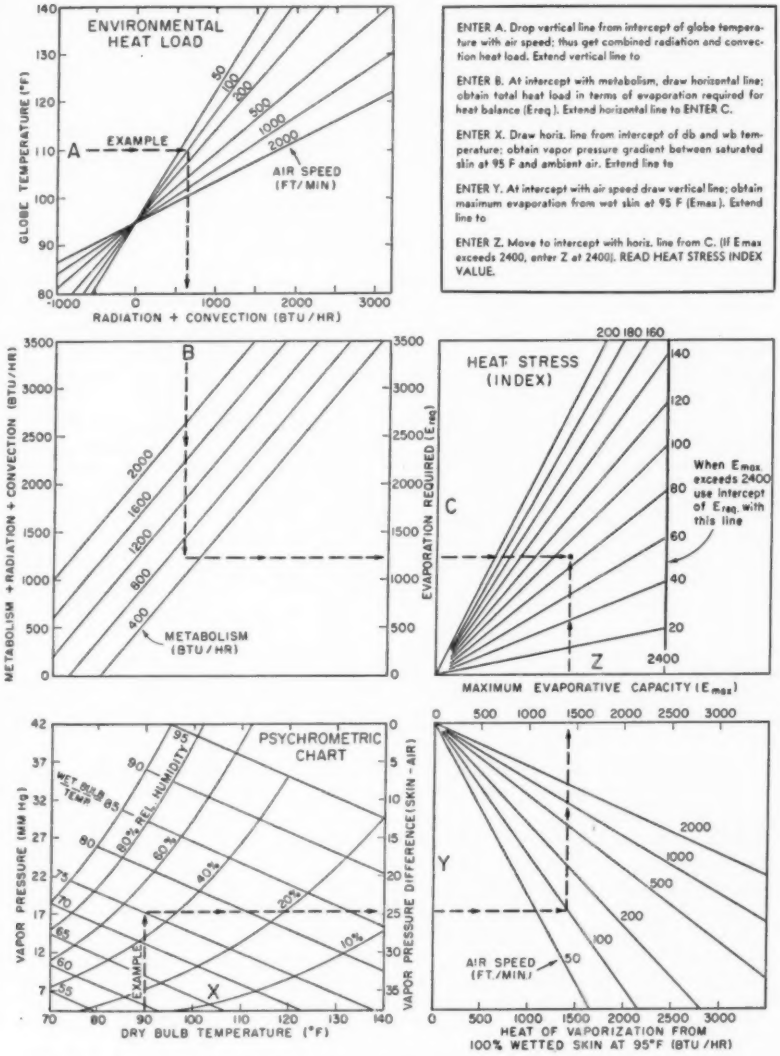
ENVIRONMENTAL HEAT LOAD
GLOBE TEMPERATURE (°F)
RADIATION + CONVECTION (BTU/HR)
EXAMPLE
AIR SPEED (FT/MIN)
50
100
200
500
1000
2000
A
ENTER A. Drop vertical line from intercept of globe temperature with air speed; thus get combined radiation and convection heat load. Extend vertical line to
ENTER B. At intercept with metabolism, draw horizontal line; obtain total heat load in terms of evaporation required for heat balance (Ereq). Extend horizontal line to ENTER C.
ENTER X. Draw horiz. line from intercept of db and wb temperature; obtain vapor pressure gradient between saturated skin at 95 F and ambient air. Extend line to
ENTER Y. At intercept with air speed draw vertical line; obtain maximum evaporation from wet skin at 95 F (Emax). Extend line to
ENTER Z. Move to intercept with horiz. line from C. (If Emax exceeds 2400, enter Z at 2400). READ HEAT STRESS INDEX VALUE.
METABOLISM + RADIATION + CONVECTION (BTU/HR)
EVAPORATION REQUIRED (Ereq)
B
2000
1600
1200
800
400
METABOLISM (BTU/HR)
HEAT STRESS (INDEX)
C
When Emax exceeds 2400 use intercept of Ereq with this line
Z
2400
MAXIMUM EVAPORATIVE CAPACITY (Emax)
PSYCHROMETRIC CHART
VAPOR PRESSURE (MM Hg)
VAPOR PRESSURE DIFFERENCE (SKIN - AIR)
WET BULB TEMP
80% REL. HUMIDITY
60%
40%
20%
10%
EXAMPLE
X
DRY BULB TEMPERATURE (°F)
Y
HEAT OF VAPORIZATION FROM 100% WETTED SKIN AT 95°F (BTU/HR)

FIG. 1—FLOW CHARTS FOR DETERMINING HEAT STRESS INDEX VALUES

Example: Globe 110, dry bulb 90, wet bulb 75, air speed 100, metabolism 600 (light arm work standing at bench). For solution follow broken lines: Heat stress = 90.

If a man were an automaton with unlimited sweat-producing capacity, it would now be reasonable to specify that heat stress of a situation is proportional to E_{req} up to the point where E_{req} exceeds E_{max}. It would then follow that beyond this point the stress is excessive in proportion to the amount by which E_{req} exceeds E_{max}. This criterion of stress is not justified for the simple reason that man is not a sweating automaton[6]. He does within wide limits produce sweat in an amount appropriate for maintaining approximate heat balance. But his capacity for sweating is limited, and the cooling efficiency of the sweat that is produced is known to decrease as E_{req} approaches E_{max}.

Evaluation of Heat Strain

The next part of the problem is to find what relationship exists between physiological heat strain, and values for E_{req} and E_{max}.

While there are many manifestations of heat strain, including thirst, increased blood volume, sensation of fatigue or faintness, and rising body temperature, most are dependent on the two primary strains which are directly operative for removal of heat.

The first primary strain resulting from heat stress is marked increase in circulation of blood to the skin. This brings internal heat to the skin; it also raises skin temperature, thereby increasing the temperature (and potential vapor pressure) gradient between skin and the environment. Secondary manifestations of this strain are increased heart rate, flushing of the skin, faintness and perhaps lassitude, any or all of which may be related to the diversion of blood supply from the body core to the skin. Evidence exists that for fit persons this strain is limiting only when the sweat that is secreted cannot be evaporated. For persons with impaired cardiovascular function, however, this strain will frequently determine endurance for heat.

The second primary strain is the production of sweat. To the extent that such sweat is evaporated (with the removal of approximately 1050 Btu per lb) this is a useful physiological response. Some secondary manifestations of the operation of the sweating mechanism are thirst, salt depletion (sometimes leading to cramps), and heat rash.

On the basis of these considerations, it is not surprising that heart rate, sweat rate, rectal temperature and skin temperature show significant correlations with environmental heat stress. Increased heart rate and skin temperature† are indirect reflections of the first primary strain. Both of these and rectal temperature may be indirect manifestations of failure to maintain bodily heat balance, as a consequence of high heat stress. But *sweat rate is itself one of the primary manifestations of heat strain*, as indicated above. In view both of the direct role of sweat in meeting E_{req}, and of the effectiveness with which body temperature is maintained at a *normal* level during exposure to heat, it is not surprising that sweating shows the highest separate correlation with presumed stress, as measured by Effective Temperature□. This point has been made by Wyndham et al[7].

The assumptions on which this heat stress index is constructed are: (1) that sweat rate is an adequate index of overall heat strain; (2) that sweating is proportional

† Skin temperature is also affected by the net rate of heat *removal*, by radiation, convection and evaporation.

□ Combinations of these interrelated factors yield higher correlations. Robinson[8] has proposed an *Index of Physiological Effect* which combines all four of the manifestations of strain that have just been considered.

to imposed heat stress; and therefore (3) that heat strain can be represented in terms of heat stress. There is precedent for the first assumption in that the sweat rate was accepted as evidence of strain by McArdle et al in developing P4SR[1].

Index of Heat Strain: The tasks remaining are (1) to provide an index scale which relates sweating to endurance for heat, and (2) to combine the two items of stress, E_{req} and E_{max}, in a manner which is systematically related to observed sweat rates.

Upon initial examination, the first task seemed impossible, because endurance of individuals differs with their structural attributes and physiological capacities. However, this difficulty is circumvented by arbitrarily designating a *standard man* to whom the scale will apply. This *standard man* is a young man of above average physique and fitness and with dimensions which agree with those used for the stress analysis (*e.g.*, surface area 20 sq ft, height 5 ft 8 in. and weight 154 lb); he is acclimatized to heat. This specification of the *standard man* probably represents as good a general description as can be provided of the subjects used in most laboratory and field studies of human responses to heat.

The index is to be based on sweating of this *standard man*. The index values will be most readily understood if they represent a percentage of his sweating capacity. However, this requires setting a value for sweating capacity of the *standard man*, and thereby introduces a new difficulty, because sweating capacity depends on duration of exposure. This is indicated by representative average values for maximum sweat rate taken from the literature[6] and plotted in Fig. 2. Apparently, some men can sweat at a rate of well above two liters per hour for brief periods, and between 1½ and 2 liters for four hours, but over a 24-hr period the average rate has not exceeded about half a liter per hour.

The most useful period upon which to specify sweating capacity would seem to be the normal work day of 8 hr, and a good approximation of capacity for that per-

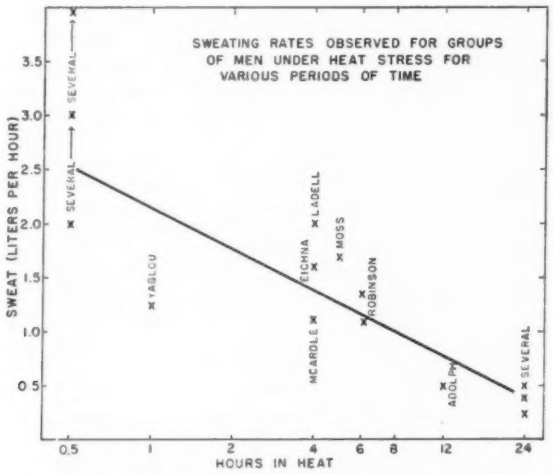

FIG. 2—AVERAGE SWEATING RATES OF MEN UNDER HEAT STRESS FOR VARIOUS PERIODS OF TIME

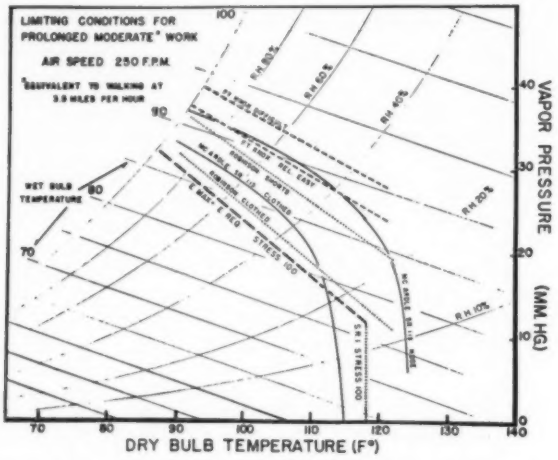

FIG. 3—LIMITING CONDITIONS FOR PROLONGED MODERATE WORK

iod seems to be 1 liter of sweat per hour (Fig. 2). *The index value* 100 *is there fore specified to represent a sweat rate of* 1 *liter per hour.* Evaporation of a liter of sweat requires about 2400 Btu.

Meaning of Heat Strain Index in Terms of Heat Stress: The final task is to combine the two factors of heat stress, E_{req} and E_{max}, in such a way that they correspond with observed sweating for a standard man.

Heat Stress 100: As a preliminary step, extensive physiological data which represent *tolerable* maximum strains for 4 to 6 hr were collected. Several sets of comparable data were found; the most extensive, for exposures of men walking at 3.5 mph, have been plotted on a psychrometric chart as Fig 3. (The bases for the lines representing these data are summarized in Table 2.) While the lines do not superimpose, they do designate a physiologically critical band of temperatures for prolonged work of this grade with this air movement*.

In terms of environmental stress and physiological knowledge, what rational basis can be found for these limits? In the first place, for any prolonged task it may be assumed that the participant must maintain or nearly maintain heat balance, because any large elevation of body temperature (more than a degree or two above normal for any activity) may be expected to result in a decrement in performance. This means that the environmental conditions must permit evaporation of whatever sweat is required to maintain heat balance; *i.e.*, E_{req} may equal, but must not exceed, E_{max}. The heavy broken line of Fig. 3 indicates, for the applicable conditions of work and air movement, the locus of points where E_{req} is predicted to just equal E_{max}. The actual value of $E_{req} = E_{max}$ increases as the vapor pres-

* The *difficult* line from the Fort Knox studies may well be judged not comparable on the basis that half of the subjects were just able to complete four hours under this condition. If this line is disregarded, the limiting conditions for prolonged work are more narrowly defined.

sure (or relative humidity) decreases, from about 800 Btu at 87 F and vapor pressure 32 mm Hg (100 percent RH) to 3400 Btu at 136 F and zero vapor pressure (0 percent RH).

However, in line with previous considerations, an E_{req} value of 2400 (1 liter of sweat per hour) must not be exceeded because this value represents sweating capacity of the *standard man* for an 8-hr period. For this reason the limit line for these particular conditions becomes fixed at a dry-bulb temperature of 118 F (dotted line) and is not continued to intercept at 136 F. McArdle's predictions lend support to the physiological postulate that when E_{max} is not limiting, the heat load, E_{req}, becomes determining.

TABLE 2—KEY TO FIG. 3

(*Limiting Conditions for Prolonged Moderate Work*)

DESIGNATION IN FIG. 3	CONDITIONS	REFERENCE NO.
ROBINSON SHORTS	Walking on treadmill dressed only in shorts; metabolic rate 188 calories per (square meter) (hour); air movement 55 m/min. These represent the most severe environmental conditions in which men maintained thermal equilibrium from second through sixth hours of exposure.	8
ROBINSON CLOTHED	Ditto above, but metabolic rate 189 calories per (square meter) (hour). Subject dressed in herringbone twill jungle uniform.	8
FT. KNOX REL. EASY	Continuous marching for 4 hr at 3 mph with 20 lb pack; (300 calories per hour) no clothes except shoes and socks. Men finished *fit to fight* with average skin temperatures 95.5 to 100.2 F; rectal temperature 100.1 to 101.2 F; and heart rate 121 to 150 beats per min.	9
FT. KNOX DIFFICULT	Ditto above; the less fit subjects were barely able to finish the four hours.	9
MCARDLE S.R. 1.13 NUDE	Based on extensive hot room studies, the curve shows predicted conditions under which a nude man will sweat 4.5 liters in 4 hr of exposure (1.13 liters per hour). This is the highest sweat rate at which every subject was able to complete 4 hr; metabolic rate 165 calories per (square meter) (hour); men nude.	1
MCARDLE S.R. 1.13 CLOTHED	Ditto above, but prediction is for men wearing overalls over underwear shorts.	1
$E_{max} = E_{req}$ STRESS 100 (BROKEN LINE)	At all points on this line, evaporation rate from wet skin at 95 F with air speed 250 fpm (from Haines and Hatch, Fig. 9) is just equal to the total heat load arising from metabolism, radiation and convection (Haines & Hatch, Table 1, Figs 3 and 6.)	4
SR 1 L. STRESS 100	This denotes conditions under which heat balance may be maintained with a sweat rate of 1 liter per hour.	This Communication

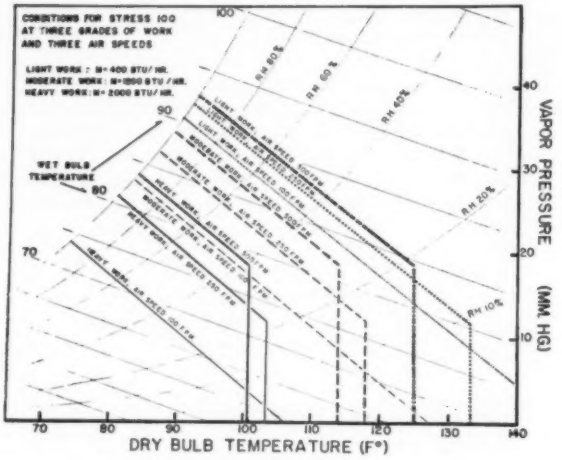

FIG. 4—CONDITIONS FOR STRESS 100 AT THREE GRADES OF WORK AND THREE AIR SPEEDS

The broken line indicating $E_{req} = E_{max}$ does not superimpose on the observed limits, but calculations show that it lies below the other lines only because a skin temperature of 95 F was assumed instead of the 97 or 98 F usually observed by physiologists under such conditions. Examination of the limit lines of Robinson and of McArdle and associates for men clothed in coveralls indicates an even better correspondence with the theoretical boundary lines derived on the basis of these physiological assumptions.

Data have been plotted here for only one grade of work and one wind velocity, but investigation using this same approach has been extended to conditions of light work and of rest. While the physiological data on these is not as extensive as for the grade of work treated here, the relationship of predicted limits to actual data on tolerance is generally similar to that just presented for moderate work. (The actual values are, of course, quite different.)

For this index, the limit lines calculated on the basis of stress are accepted as consistent with a probable strain of 100 for the *standard man*, despite the fact that the limits indicated by physiological investigation on nude men lie somewhat higher. The physiological experiments represented results of 4 to 6 hr exposures rather than the 8 hr specified for the *standard man*, and civilized man usually wears a light covering of clothing, the net effect of which usually is some increase in heat strain.

In summary, a heat stress value of 100 is assigned when $E_{req} = E_{max}$, or when E_{req} equals 2400 Btu per hr. The positions of stress-100 lines for three different grades of activity and at three air speeds are shown in Fig 4.

Heat Stress Less Than 100: In the first tentative proposal of a heat stress index by the authors[10], stress values less than 100 were assigned on the basis of E_{req} (*i.e.*, $E_{req}/2400 \times 100$ = stress) up to the point where $E_{req} = E_{max}$ (*i.e.*, index 100 as defined above). However, it was found that this procedure failed to take into

account the physiological observation that the evaporation limits often are a factor in strain even when E_{req} lies below E_{max}. This is indicated in Fig. 5 by the slope of McArdle's[1] lines for predicted sweat and Robinson's[8] for observed sweat at levels of 0.25, 0.50 and 0.75 liters per hour. In physiological terms, as E_{req} approaches E_{max} more sweat is secreted than is theoretically needed for heat balance. This does not represent *oversweating*, because the extra sweat only suffices to maintain heat balance; rather it must be regarded as a progressive inefficiency of evaporation. The reason for this is not clear, but since the cooling is not that predicted for complete evaporation of all of the formed sweat at the skin, it may be assumed that some drips off due to inefficient distribution of sweating activity over the body surface, or that the sweat is evaporated at some distance from the skin (as from the top of droplets). Gagge[11] introduced a concept of *wetted area* in relation to this inefficiency and accompanying sensation of discomfort.

A simple way of improving the fit of the lines of equal stress with observed sweat rates is to consider the stress proportional to the ratio between E_{req} and E_{max} up to the point where this ratio is less than that indicated by E_{req} alone. The heavy broken lines of Fig. 5 have been constructed on this basis:

$$(E_{req}/E_{max}) \times 100 = \text{index} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

whereas the dotted lines indicate stress determined on the basis of the maximum sweating capacity.

The higher index values constructed on this basis lie below the corresponding observations on nude men apparently for the same reason that they did for stress 100 (*i.e.*, skin temperature of 95 F is used in computing the stress). The lower index values correspond better, perhaps because the 95 F skin temperature assumed in computing the stress is close to that actually observed under such conditions of exposure.

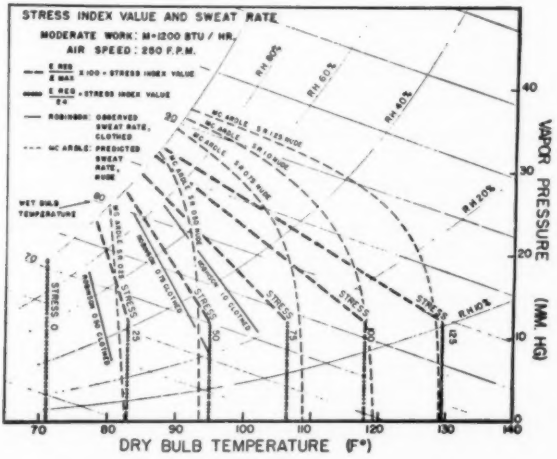

FIG. 5—STRESS INDEX VALUE AND SWEAT RATE

On the basis of the above considerations, heat stress index values are computed on the basis of $(E_{req}/E_{max}) \times 100$ whenever this yields a higher value than that obtained by using $(E_{req}/2400) \times 100$.

Heat Stress Greater Than 100: The same system may be used for determining and stating stresses greater than 100. Here, when E_{max} is less than 2400, the index is $(E_{req}/E_{max}) \times 100$. When E_{max} exceeds 2400 the index is $(E_{req}/2400) \times 100$.

When heat stress exceeds 100 the implication is that heat balance may not be maintained and that the sweating mechanism may be overtaxed, so that the *standard man* will not tolerate prolonged exposure. Such conditions are actually found in some industrial jobs and during hard work in hot weather. The overall stress under such circumstances will be indicated by the stress index, and the maximum endurance of a single exposure will depend on the worker's tolerance to elevated body temperature. On the basis of physiological studies, it seems likely that the *standard man* may safely tolerate an elevation of body temperature of 2 F, and for him this represents an accumulation in the body of about 250 Btu of stored heat. His maximum tolerance time for a single exposure might then be calculated by using Equation 8:

$$250/(E_{req} - x) = \text{maximum allowable exposure in hours} \quad . \quad . \quad . \quad . \quad (8)$$

where

x = 2400 or E_{max}, whichever is smaller.

Minimum allowable time for recovery depends on the amount of heat stored in the body as well as on environmental conditions and bodily activity during the recovery period. On these bases, the minimum time (in hours) for recovery might be represented by:

$$\frac{H_{stored}}{H_{sweat} - E_{req\ recov}}$$

in which

H_{stored} = amount of heat stored.
H_{sweat} = maximum heat removal by sweating.
$E_{req\ recov}$ = E_{req} of recovery environment.

Maximum heat removal by sweating might be given the value 2400, or E_{max} of the recovery environment, whichever is less. E_{req} *of recovery environment* might, of course, be a negative figure if the environmental conditions were cool enough.

Since there have been few physiological studies of tolerance times and of the recovery process within the stress range that is of interest, the validity of these procedures is questionable.

Determination of Heat Stress (*Index Value*): Fig. 1 has already been introduced as a method for evaluating heat load (charts A and B) in terms of E_{req}, and of maximum evaporative capacity (charts X and Y), E_{max}. If the remaining chart is now entered at C and Z the intercept will indicate the applicable heat stress index value.

Examples: Work sheets applicable to three different jobs are summarized in Table 3. The first two have been somewhat simplified in that radiant and convective heat loads have been considered the same in summer and winter. In reality the air temperature is somewhat lower in winter and higher in summer than the value given, but the significantly lower stress in winter is largely accounted for by the lower vapor pressure and consequent greater evaporative capacity of the air.

TABLE 3—ANALYSIS OF HEAT STRESS OF THREE JOBS IN TERMS OF PHYSIOLOGICAL STRAIN

DESCRIPTION OF JOB	JOB #1 TRANSFER OPERATOR AT GLASS FORMING MACHINE	JOB #2 INSPECTOR AT GLASS FORMING MACHINE	JOB #3 MINERS; DEEP MINE
Globe temperature, F deg	130	104	93
Air temperature, F deg	100	90	88
Wet-bulb temperature (F deg) Summer	80	75	88
Wet-bulb temperature (F deg) Winter	60	58	88
Air speed (ft/min)	200	200	200
Radiation and Convection (Btu/hr)	1800	500	−100
Metabolism (Btu/hr)	1000	600	700[a]
E_{req} (Btu/hr)	2800	1100	650
E_{max} (Btu/hr) Summer	1700	1800	650
E_{max} (Btu/hr) Winter	2900	2900	650
Raw Index of heat stress Summer	175	60	100
Raw Index of heat stress Winter	120	45	100
Percent time in heat	50	80	100
Index of total job heat stress Summer	88	48	100
Index of total job heat stress Winter	60	36	100

[a] Includes rest pauses; hypothetical case constructed from Wyndham et al[11].

The first example cited represents a very severe exposure to radiant heat where two men customarily work in rotation on the job.

The second example represents a job requiring little physical effort; in some jobs of this type a 10- or 12-min rest break is allowed each hour.

The third example, that of the miner, is partly hypothetical and represents about the worst conditions at which physiologists have found that acclimatized laborers can work steadily at a moderate pace[12]. Without extensive supply of fresh air by forced ventilation, the heat stress would be much worse because the walls would then attain a temperature as high as 120 F, and the air temperature would approach that of the walls; such conditions cannot be tolerated long even by resting men.

If exposures are intermittent or a job is varied, there may be justification for averaging the stresses (*e.g.*, Table 3). For example, a man in a glass factory works under a stress of 70 for 3 hr, 20 for 2 hr, and escapes completely from exposure for one hour; it may be fair to express his overall heat stress as

$$[(70 \times 3) + (20 \times 2)]/6 \leqq 42 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Since the standard 6-hr day in the glass industry represents recognition of exposure to heat stress, it might even be considered legitimate to use 8 hr in the denominator, which would yield a daily stress average of 31. The caution in interpretation in this case derives from the fact that some persons able to tolerate a 31 stress if it represented 8 hr of light work might not be able to tolerate the 3 hr of stress 70. This might be particularly true if the 70 exposure represented heavy muscular work,

TABLE 4—EVALUATION OF INDEX OF HEAT STRESS

INDEX OF HEAT STRESS	PHYSIOLOGICAL AND HYGIENIC IMPLICATIONS OF 8-HR EXPOSURES TO VARIOUS HEAT STRESSES
−20	Mild cold strain. This condition frequently exists in areas where men recover from exposure to heat.
−10	
0	No thermal strain.
+10	Mild to moderate heat strain. Where a job involves higher intellectual functions, dexterity, or alertness, subtle to substantial decrements in performance may be expected. In performance of heavy physical work, little decrement expected unless ability of individuals to perform such work under no thermal stress is marginal.
20	
30	
40	Severe heat strain, involving a threat to health unless men are physically fit. Break-in period required for men not previously acclimatized. Some decrement in performance of physical work is to be expected. Medical selection of personnel desirable because these conditions are unsuitable for those with cardiovascular or respiratory impairment or with chronic dermatitis. These working conditions are also unsuitable for activities requiring sustained mental effort.
50	
60	
70	Very severe heat strain. Only a small percentage of the population may be expected to qualify for this work. Personnel should be selected (*a*) by medical examination, and (*b*) by trial on the job (after acclimatization). Special measures are needed to assure adequate water and salt intake. Amelioration of working conditions by any feasible means is highly desirable, and may be expected to decrease the health hazard while increasing efficiency on the job. Slight "indisposition" which in most jobs would be insufficient to affect performance may render workers unfit for this exposure.
80	
90	
100	The maximum strain tolerated daily by fit, acclimatized young men.

because the heat production, but not the accompanying physical strain, of work is represented by the index.

Significance of Different Index Values: While the principal purpose of this communication is to propose a means for evaluating stress in terms of the *standard man*, rather than to discuss the strains of exposures on people with different physical attributes, some interpretation of the stress figures in terms of strains on workers seems desirable. Table 4 represents an attempt to summarize some implications of daily exposures to various heat stresses in terms of present general understanding of heat effects.

Interpretations must be made with caution. If the subject is a woman or child, is elderly, suffers circulatory or respiratory impairment, or is not acclimatized, the index values have an absolute meaning different from that for the *standard man* even though the relative meaning may be the same. Heat exposures result in more strain on large and overweight persons, basically because the amount of their skin surface (through which the heat must be lost) is relatively small in relation to their mass (which produces the heat).

The competence of different individuals to withstand any type of stress undoubtedly follows the *normal distribution curve*. If a required production of 1 liter of sweat per hour for 6 or 8 hr a day on several days a week is just tolerable for our young, physically above average, acclimatized *standard man*, it seems likely that no more than 10 to 20 percent of the working population could be expected to

work successfully under a stress of 100. It is also reasonable to suppose that almost all of the working population could successfully withstand strains given by index rating 10, 20 and 30, and termed *moderate*. A majority of the population should withstand 40, 50 and 60, the stresses that have been categorized *severe*. It is notable that the ability to withstand heat stress decreases with age, and that this may be related to progressive impairment of cardiovascular function. Since cardiovascular functional capabilities of women are less, even for their size, and since there is somewhat greater insulation in the skin of women it seems probable that the heat strain for women corresponding to any given stress is somewhat more severe than is indicated by this index. However, these points require verification.

There is considerable evidence that prolonged mild heat stress (Indexes 10 to 30) affects alertness and drive, and may subtly affect performance of higher intellectual functions. On the other hand, there seems to be little decrement in the performance of moderate to heavy physical work under such conditions unless the ability to perform such work is marginal under conditions of no thermal stress. Conditions of prolonged *severe* heat stress (Indexes 40 to 60) seem unsuited for activities requiring sustained mental effort, but may not affect physical effort of men who are physically fit and have been acclimatized. Under conditions of prolonged *very severe* heat stress (Indexes 70 to 100) performance of physical work may be markedly impaired even though the men involved suffer no obvious deleterious after-effects from exposures.

APPLICATIONS OF STRESS INDEX TO PRACTICAL PROBLEMS

It is believed that the method described here is unique in that it yields a single figure descriptive of the overall heat stress involved in an exposure—a figure which is quantitatively comparable with similarly derived index figures for other exposure conditions.

The index may be of value wherever there is need to describe hot conditions and wherever there are problems related to heat exposure. The climatologist and geographer (and even the U. S. Weather Bureau) might find it useful to describe hot weather; the only need would be to fix a *standard activity* for the *standard man*.

Air conditioning engineers, military command and medical personnel, industrial hygienists, methods engineers and physicians may wish to use it on such problems as: (1) regulating work of men in hot climates; (2) predicting heat danger to those engaged in out-of-door tasks where solar radiation is a factor; (3) selection and assignment of persons fit to withstand heat stress; (4) reaching decisions regarding the relative need for remedial action, either to improve an environment or lighten the work required in an environment; (5) determining on a rational basis maximum allowable time of single exposures; (6) specifying hygienically desirable conditions for recovery following exposure to heat; (7) surveys of populational exposure to heat; (8) providing a basis for arbitration of pay bonuses for hot jobs.

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DISCUSSION

L. A. BROUHA†, Wilmington, Del. (WRITTEN): As indicated by the authors, their aim is to provide an index of heat stress based on physiological strains predicted to result from exposure. They have approached this complicated and important problem in a logical way, attempting (1) to provide a means for evaluating the components of heat stress; (2) to determine the relation between these components and the resulting physiological strains; and (3) to assign to the stresses index values which have meaning in terms of human endurance for heat.

The evaluation of components of heat stress is sound and simple to derive from the formula:

Metabolism + Radiation + Convection = Evaporation Required, using Fig. 1, Chart B.

Nevertheless, one must remember, as pointed out by the authors, that this way of figuring heat load assumes a fixed body surface area, a fixed skin temperature and a no clothing basis. These are 3 important variables when dealing with groups of industrial workers, and it is admitted that "estimates of load may be seriously in error if heavy clothing is worn" which is the case for many jobs in heavy industries. The same remark applies to the assumptions made in estimating maximum evaporative capacity. Consequently the values of E_{req} and E_{max} can only be approximations.

Assuming that these approximations are accurate enough for general purposes, the valuation of heat strain is well presented as applied to the *Standard Man* and his approximate sweating capacity of 1 liter of sweat per hour for a normal work day of 8 hours. This leads logically to a heat stress value of 100, assigned when $E_{req} = E_{max}$ or when E_{req} = 2400 Btu = 1 liter of sweat per hour for an 8-hr period. Evaluating the stress as being proportional to the ratio between E_{req} and E_{max} up to the point where this ratio is less than that indicated by $E_{req}/2400$ seems justified.

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† Physiologist, Haskell Laboratory for Toxicology and Industrial Medicine, E. I. DuPont de Nemours and Company.

In the last part of their paper the authors give good examples of evaluating the index of heat stress and discuss objectively some of the limitations of their method. Age, sex, and physical strain due to heavy muscular work are mentioned as points that require verification and justify the statement that "interpretations must be made with caution."

It has been our experience over the last 12 years that workers who are in good physical condition, fully acclimatized, exposed to the same heat stress and carrying the same work load show a wide range of physiological strain when their reactions are actually measured *on the job.* It should also be mentioned that there are many jobs in which the heat stress, the air speed, and the work load are not constant throughout the 8-hr shift. They may vary considerably according to the nature of the various operations in which the workers are involved. As mentioned by the authors, "there have been few physiological studies of tolerance times and of the recovery process within the stress range that is of interest, . . ." This is obviously the next step to be taken and more data are needed to establish accurately the relation between the heat stress index and the range of physiological strain as actually measured in various industrial situations.

Belding and Hatch deserve full credit for establishing a single figure index of heat stress which takes into account physiological factors of strain, which is simple and which, if properly handled, should give valuable information to determine acceptable work load and heat stress as well as adequate rest periods in *hot jobs.*

K. L. DUNN*, Corning, N. Y., (WRITTEN): It is indeed a privilege to be asked to discuss this very excellent paper because I am in the somewhat rare position of having had the opportunity to witness and discuss many of the developmental phases of the studies leading to this presentation.

As one who is required in day by day work to deal with environmental situations in a high temperature industry, I can truthfully state that the problem of heat as it affects the employees and methods to alleviate these conditions has been in a rather vague state for a good many years. The only corrective procedures used up until about 12 years ago was the same method the average layman might use when weather becomes hot. When an employee became warm on the job and he began to complain, the procedure was to provide fans. As the weather became still hotter the answer was to install more and more fans.

It was only about 10 or 12 years ago that one of the coauthors of this paper began to think analytically on the problem of industrial heat and at a discussion at one of our plants suggested that while we were doing an excellent job of correcting convective heat loads on our employees with the "fan and more fan" philosophy, very little was being done about radiant heat.

After several more informal discussions not only in my own company, but in other glass companies, a joint study was set up under the auspices of the Industrial Hygiene Foundation to investigate and formulate methods of measurement of the various phases of the heat problem in industry as it affects employees and attempt some beginning procedures on methods of control. The work of Vernon in England using the globe thermometer and various studies by the United States Bureau of Standards were reviewed and with certain modifications adapted as means of measurement in the initial studies. The summary of this study along with the description of procedures was reported by Haines and Hatch in *Heating and Ventilating* in November 1952.

Out of all of this preliminary work came the realization to those of us in industry who had been following these studies that the big problem in the high temperature trades, at least, was one of radiation. We could not at the same time, however, disregard the other factors of convection and evaporation but we did attack vigorously the previously unrecognized radiant heat problem. This phase of correcting industrial environmental temperatures we could easily dub the "Aluminum Shielding" phase. The shielding of high temperature sources progressed so rapidly that it came near to getting out of hand;

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in some cases even to the point of introducing a misconception that the mere hanging of aluminium or aluminium foil in a hot work space would produce magical refrigerating qualities. One of our colleagues in the glass industry had the experience of employees insisting upon his hanging globe thermometers about the work space to cool it down!

It should be understood that during this initial phase of practical application of this joint study most of us in the field were far from being theoretical physicists or physiologists and for those of you who have attempted to struggle with the intricacies of the Stefan-Boltzmann equation and other equally complicated formulae for energy transfer, convective heat and air movement, will agree that such procedures are hardly in the realm of the practical industrial hygienist or plant engineer. The mere gathering together of the M plus R plus C of Equation 1 of the present paper began to assume the magnitude, to most of us, of moving the Empire State Building six blocks up town.

You will note from this remark that there were a few of us in the practical field, along about this time, who had begun to realize that the crux of the heat problem was not one of pure physics, but what we were after was an attempt to correct an unpleasant or dangerous condition in a work space. The word metabolism at this stage of the work had the simple meaning to most of us in the industrial world, of a gadget that one was hooked to to determine whether one had a goiter or not. With the introduction of this term *metabolism* and several others that have been mentioned in the present paper it became evident that if we were really to make these new tools of measurement work best we were going to have to find some means of proving that in addition to feelings of comfort we had actually accomplished something that could be measured in terms of those feelings.

The simplest way to have such a figure presented, it seemed to some of us, would be to have a simple linear graph with globe thermometer temperatures down one axis and maximum allowable work time along the other and merely use the intercept of such a simple line to determine whether things were too hot or too cold in the work space under survey. This request for such a simple graph was made rather facetiously several times to both authors of this present paper and in nearly every instance the only answer to our request for such a linear chart was an enigmatic smile on the part of both. Very frankly our first view of their "Flow Chart Method of Determining Heat Stress Index Values" as illustrated in their Fig. 1 was extremely startling. My first reaction was that my leg was being pulled until I had an opportunity to try the charts in practice. Before I go further I would like to state I think the chart, as presented by the two men, is one of the most valuable contributions to practical applied industrial hygiene that has come to us in many years. I can only hope that this chart and the material contained in the paper will become a part of standard engineering and industrial hygiene textbooks and handbooks. I might go further and say that I would hope that someone would produce the chart in pocket size, sealed in one of these plastic envelope type of cards that one could carry with him in industrial hygiene surveys.

While the authors caution several times during the paper that some of the figures are untried to a great degree I will state at this early date that where we have had the opportunity to try this method and apply the material found in this paper; we have had a remarkable correlation with employee opinion as to the working conditions to which he is subjected.

In the author's discussion of the applications of the stress index to practical problems we have had in my laboratory greatest use of this method in determining percentage time that one could reasonably work in hot weather at given jobs and also to calculate optimum conditions for recovery areas for extremely hot jobs. While I do not hold personally with the philosophy of paying a bonus for working in an uncomfortable or dangerous environment, we have used this method to arbitrate between 2 separate groups of employees as to which was being exposed to the more uncomfortable conditions when differences of opinion have arisen because of pay differentials between 2 different jobs. Those of us who work with the hot trades can see almost endless possibilities in the applications of the methods outlined in this present paper. I can only suggest to other people practicing in the field that they should immediately put this method to use, not

only in hot weather but in cold weather to accumulate much needed data to verify these present data or to allow the authors to correct, where corrections are or may be needed.

And now in closing it would not be fitting if I did not at least offer some constructive criticism of this presentation. It is a criticism that I believe can easily be corrected and that is the business of estimating energy metabolism for various types of activity. We have found that there are those of us that can make a reasonably accurate estimate as to whether a man is doing work equivalent to sitting, standing, or the various categories that are present in most standard energy metabolism tables, such as was illustrated in Table 1 of the present paper. But we would visualize a much more extensive table in greater detail utilizing material that may well be on file in the physiology laboratories, incorporating such material into a table similar to that which we find in illumination problems, wherein a given job is described quite minutely and a foot candle requirement placed in the right hand column. It would seem that a more complete breakdown could be made in energy metabolism tables; such as the Btu per hr required to saw off a standard 2 x 4 in. pine plank in 15 seconds, or the energy requirement of shoveling sand from the ground to a 3 ft high platform at the rate of 100 pounds per minute. Details such as this would be of great assistance to the uninformed, as we were, in estimating energy expenditures in unmeasured operations such as are found in the glass industry. Perhaps some of this needed data could be assembled by on-the-job determinations, in industry, by means of the Douglas Bag method of calorimetry using normally clothed individuals who are accustomed to doing the job being measured. The one shortcoming, if it is that, in applying this new tool in industry is our clumsiness in projecting the work activity at hand to a "man, naked, carrying a fifty pound pack on a treadmill walking 3 miles per hour."

As stated, however, in reasonably skillful hands we have made successful use of the method described and feel assured that our slight discomfort in translating physiological data can be alleviated.

C. P. YAGLOU, Boston, Mass. (WRITTEN): The heat stress-strain index proposed by Belding and Hatch essentially is an elaboration of a rational concept originally proposed by the Climatology and Environmental Protection Section, Office of the Quartermaster General, in the development of the *Thermal Acceptance Ratio.* Basically, the Belding-Hatch index is the reciprocal of this ratio, utilizing the sweat rate, instead of the metabolic rate, as a criterion of heat strain. An important forerunner of both of these schemes is Robinson's *Index of Physiological Effect* based on a combination of the 4 principal heat strain criteria, i.e. heart rate, skin temperature, rectal temperature, and sweat rate, weighed equally.

The most hopeful possibility in these new approaches lies in evaluating the thermal loads, not in terms of physical environmental factors as in indexes now in use, but in terms of physiological strain, or the actual cost to the body imposed by these loads. Although this approach is admittedly complex, it offers a rational solution to the assessment of thermal environments.

Reliable information is not now available for estimating all of the components of heat stresses, or for assessing the relative importance of heat strain criteria to be used in different situations. In the absence of this information, a number of assumptions and limitations must be resorted to, as pointed out by the authors, hoping that the positive errors will tend to counteract the negative ones. Caution should therefore be exercised in the use of any new and untried index, until its practicability and limitations have been demonstrated in the field. The use of 2 or more indexes is always safer than of any one alone.

In this reviewers opinion, the only serious exception one might take to this presentation concerns the premature claims made for this new index at the end of the paper.

C. L. TAYLOR‡, Los Angeles, Calif. (WRITTEN): These authors are to be congratulated on their formulation of this complex problem. Necessarily, many approximations had

‡ University of California at Los Angeles.

to be made in order to reduce the analysis to a manageable form. In the main this task has been well done. It is particularly commendable that they have combined the precision of heat transfer analysis, with the realistic, even though empirical, results of human exposures to heat loads. Experience in many fields has shown that such a biotechnical method provides a form of expression that is applicable in the practical situation, yet faithful to the unanalyzed aspects of human response. Moreover, it provides a formulation of heat tolerance which can be accepted by the engineer, the physiologist and the industrial physician.

The general framework of the analysis, based upon heat exchange equations suitably evaluated by empirical determinations on human subjects, has become conventional in analyses of this type; however, there are some new features which call for comment:

(1) The reintroduction of the globe thermometer (following Hatch and Haines, *Heat. and Vent.* Nov. 1952) is noteworthy. While readings with this instrument do not exactly parallel the convective and radiant exchange of man, this discrepancy is far outweighed by the common neglect of radiation measurement, which seems to be largely due to the lack of simple instrumentation for this phase of environmental assessment.

(2) The inclusion of maximum sweat rate limits in the stress index represents a healthy reaction against the over-simplified views that many have expressed, following an uncritical application of the Gagge wetted area theory of evaporative regulation. It has been clear from many investigations (cf. Robinson's and McArdle's curves in Fig. 3) that lines of equivalent effect are curved when plotted on temperature-vapor pressure charts. This means that aspects of body evaporative response have an important influence, as well as the evaporative potential of the physical environment. Thus with constant air movement, going from high to low humidity environments, there is required an increasingly higher sweat rate. Purely physical thinking would conceive that the body will meet this on the same linear relationship that governs environmental evaporative potential. The facts are otherwise, and non-linearity is introduced as the body must accept high *strains* in order to produce high sweating rates.

Belding and Hatch have met the problem of non-linearity by joining 2 straight lines: one calculated from the E_{max} equation; the other taken from the experimental data on maximum evaporative rates. (see Fig. 3, stress 100 line) Certainly the inflection point at 13 mm Hg and 118 F cannot mean a precisely abrupt transition in the physiology of temperature regulation from $E_{max} = E_{req}$ to E_{max} = a constant. Here, it might be argued that curves should be attempted rather than the two-line approximation.

(3) The statement that the index scheme is on a no-clothing basis is somewhat misleading, since the placement of the Stress 100 line is evidently determined by studies of clothed men, as against those of nude man. It is certainly true that quantitative knowledge concerning the effect of clothing upon thermal tolerance is fragmentary. This effect varies with humidity, air movement, level of sweat rate and many other factors and there is no simple way to escape the fact that we lack sufficient knowledge in this area. However, a clearer statement of the type of clothing, which is implied in the formulation of the index, would aid the practitioner to confine the index to conditions for which it is valid. Alternatively, a description of the type and amount of clothing which would invalidate the analysis might be desirable.

In conclusion, I would say that the authors have done an excellent job in reducing a complex problem to its significant parameters, and that they have met the need for a thermal tolerance index applicable to working man in the hot industries.

J. H. CLARKE, Chicago, Ill. (WRITTEN): The authors are to be congratulated on this very fine paper which covers a subject that is of growing importance to industry, the industrial ventilation engineer, industrial hygienists, and others interested in the welfare of the worker in his industrial environment. We on the TAC on Industrial Environment are happy to have had a part in sponsoring this paper. It is hoped as time goes on that this Stress Index proposed by Belding and Hatch will find the same usefulness and acceptance in industrial applications as the present Effective Temperature Chart has for comfort and some industrial air conditioning applications.

Any such index, to be a useful tool must be well understood as to the significance of its components. Because this index must incorporate so many important variables, such

as radiation loads, humidity, work rate, health, clothing, working conditions and acclimatization, it should be expected that an understanding of the significance and application of the index points will take time. In the future, designers and others concerned with evaluating the industrial environment will become more accustomed to applying the work rates of Table 1 to their problems and to interpreting the index evaluations noted in Table 4. Future research on the subject should do much to extend our knowledge on the subject and more sharply define the limits of stress indicated by Table 4. The authors point out that the heat stress index is based on healthy young men of approximately 20 sq ft skin area, 5 ft-8 in. height and about 154 lb in weight. They also note that the heat stress in small or thin men is less than for large or fat men. There appears to be a trend toward taller, heavier men and women in this country at least. If this is true, the index may require re-evaluation in the future to reflect the change.

This Society and other organizations have done extensive research on the physiological reactions of workers at various work rates and environmental conditions. A considerable amount of this work failed to evaluate the effect of radiation on the worker in an industrial environment. While the concept of body heat balance is not new, the Belding and Hatch index appears to be the first which takes the 2 very important factors of radiation and heat balance into account as primary factors of an environmental index.

There may be some criticism of the index proposed by the authors because of the fact that it is based on an average individual and certain assumed conditions. Much of the same criticism could be leveled at the Effective Temperature Chart and the M.A.C. limits accepted for toxic concentrations. And yet these measuring sticks have been very useful in providing acceptable plateaus for design. Corrections have been made to these limits as more data became available, without in any way impairing their acceptability or usefulness. This new index provides a splendid basis for a meeting of minds among industry, management, labor and the designer for establishing and maintaining equitable working conditions in the industrial environment.

W. C. L. HEMEON, Pittsburgh, Penna.: I think one point needs clarification, which I don't think the authors in their answers to the discussion have emphasized sufficiently. Attention is directed to the word *index*. This paper has presented two things, description of the index itself and the scale ranging from 100 down to zero. This is the heart of the paper. Secondly, for illustrative purposes, the authors tentatively describe a basis for interpreting the meaning of 70 or 30 or 100. The discussers, it seems to me, have offered constructive criticisms but most have been concerned with the interpretation discussion rather than with the method for constructing the scale. I would suggest that one look upon this index somewhat like the invention of the thermometer or the effective temperature index. It simply provides a basis for describing the thermal environment. Scientific differences of opinion on the physiological significance of some particular stress is another matter entirely.

W. L. FLEISHER, New York, N. Y.: I feel I have to rise in order to, in a way, criticize our chairman for apologizing for having this type of discussion brought in again after a period of years. I suppose that I have been associated with this particular phase of our industry longer than anyone here, having been chairman of almost every committee connected with environmental conditions, and I feel that it is responsible for the Society being as great as it is.

Some people think that radiation and things of that sort have been or are responsible for increased public interest in the Society, so that we have become the great institution that we are. I feel that if the TRANSACTIONS of the society are carefully analyzed, one will find there are a good many things in connection with the work that is to be carried on which should have been done not over the period of the next 5 years but over the last 5 years. The discussion in connection with the physiological conditions of the subjects entering these various test rooms has never been adequately taken care of although it has been mentioned in the records of Houghten over a good many years of laboratory testing. I mean, the rise in temperature and the ability to lose moisture of the particular subject.

In many of the cases where we tested for the point at which collapse took place we didn't take into consideration the entering temperature of the subject which may have varied as much as one and a half degrees, and those subjects, after a particular period of time, rose to a temperature which was the same as those others that were not as much affected by the conditions. In other words, we had subjects entering the room at 97 F and at 102 F. Practically all of the subjects had the same temperature, final temperature, not withstanding the fact that there was such a difference in their temperature on entering. I suggested at one time that we never gave enough effect to the loss of moisture due to the different PH concentration of the blood of people. A great many people didn't agree with me but I took it from my investigations of the effect of loaves of bread entering ovens where the PH concentration of the loaves made it possible to abstract more or less moisture. In other words, at PH concentrations of 4½ you could retain a certain amount of moisture where with a higher PH concentration, your loaves at the finish of baking showed a great deal of difference due to the difference of that particular phase.

I never was able to get the laboratory or anybody else to find out the effect of the retained moisture and the ability to retain moisture due to some physiological change in the structure of the body. I simply mention these things because I say I have been connected with this particular subject, I think, so long as Dr. McConnel is absent, longer than anybody else in the Society. I am extremely interested in everything that takes place of this nature and think it is one of the great subjects of universal appreciation.

DR. F. K. HICK, Chicago, Ill.: We have been very much interested in whether or not sweat rates at lower levels constitute any real *stress* at all. I should very much like to hear from Dr. Belding as to whether he thinks that a 30 percent index on this level really constitutes a *stress*. Certainly in 2 hours exposure we cannot demonstrate that it does so. I should like to also ask if he can define that *stress* in any measurable way in the body other than by the loss of weight.

R. S. ASH, Phoenix, Ariz.: Is the assumption made that all of the sweat evaporates from the body and if so, what effect does the run-off of sweat have upon the index?

AUTHOR'S CLOSURE (Professor Hatch): We are particularly happy to have comments from Dr. Brouha, a physiologist who is actively concerned with the heat problem in one of our important industries. He has made 3 points of particular importance. First, he calls attention to the complicating factor of clothing. There is no question about this and no question about the need for further studies to give us a better basis for incorporating the effect of clothing into the evaluation of heat stress. He, for example, is interested in workers in chemical plants who have to wear clothing for protection against chemical agents. Such protective clothing may include a vapor-proof outer garment to prevent the build-up of toxic vapors next to the skin. Such impermeable clothing will also seriously impede the outward travel of water vapor.

Recognizing this complication, we feel that a rational basis must be provided on which to examine the effect of these variables. In the absence of such a basis, it would be difficult if not impossible to relate one situation to another.

The second limiting factor noted by Dr. Brouha is the characteristic of individual variability in response to heat stress. This, too, is well recognized and we agree that a single chart, such as the *HSI* chart, will not necessarily describe the stress-strain relationship for a particular individual. It is based on the *average* response of groups of individuals. In this connection the physician must play an important part in his examination of men to determine their relative capacities (in respect to their cardiovascular capacities, for example) to cope with heat stress.

A third point in Dr. Brouha's discussion has to do with the work-recovery pattern. Given a work situation in which the heat stress is too great, this will be followed inevitably by a rise in storage of heat in the body. If, in the practical situation, it is impossible to reduce the level of exposure to an acceptable one, then the only alternative remaining to us is to regulate the periods of work and relief so that the accumulated heat load on the worker during his period of exposure is completely offset during a following

period of relief. This is common practice in industry today, but it is done on a completely empirical basis.

Given a certain load on the man in terms of environmental heat plus the metabolic heat, what is the optimum environmental condition to be maintained in the recovery area and what is the required period of stay at the relief station to offset the period of exposure at the work station, so that there will be no net accumulation of body heat during the total period? Brouha has shown that in some work situations the period of relief is not adequate to offset the period of work and, as a result, there is an accumulation of heat during the day.

An important need now is to develop a rational and quantitative basis for designing the work-relief cycle so that there will not be an accumulation of heat from one cycle to the next. Dr. Brouha is the leading investigator at the present time who is dealing with the problem in industry. It is one of the most important factors that will be looked into in the further research in our laboratory, to which Dr. Belding has referred.

Mr. Clarke's comments are very much appreciated, and especially so, because of intimate concern with the practical problem in industry. He has emphasized the need, particularly, for the development of appropriate thermal standards for industry. At the Society meeting in Philadelphia in 1951, I called attention to this need. The location of the comfort zone is rather well established. Also, we have various data such as those from Fort Knox and others which effectively represent the upper limit of human tolerance to heat. The lines of comfort and of maximum physiological tolerance are widely separated on the psychrometric chart. What we are confronted with in industry is the large area on the chart between the two limits. We need a basis for fixing heat tolerance limits for a particular industry engaged in a particular kind of work, and exposing particular kinds of people, somewhere between the limits of complete comfort and maximum physiological tolerance. We are concerned here with hot industries, in which, by definition, complete comfort is not economically attainable. The question is, how much is the industry justified in spending to minimize the heat exposure? It is our hope that the concept of the Heat Stress Index will provide a rational basis on which to fix the needed standards.

AUTHOR'S CLOSURE (Doctor Belding): I have two comments concerning Dr. Taylor's discussion. One is in regard to the sharp inflection in the line which indicates stress 100. This results from the particular assumptions that we make, and I suppose in terms of physiological responses such an inflection point could not be clearly established, partly because it is very hard to obtain precise measures of strain, and also because the responses even of one man differ from day to day. Nevertheless, using the two criteria that have been described does yield a reasonable approximation of the heat stress, and, of course, an approximation is our only concern for this index.

My second comment is that we did not mean to be misleading when we stated that the index scheme is on a *no clothing basis.* The index is dependent on the constants developed at Fort Knox on *nude* men, and used by Haines and Hatch, together with an assumed skin temperature of 95 F, for calculation of E_{req} and E_{max}. Calculations of stress made on these bases happen to coincide quite well with observed strains on men clad in cotton shirt and trousers or other light clothing. This correspondence is largely the result of the use of 95 F as a skin temperature, a value that is 2 to 4 deg F lower than that which actually obtains under high stress conditions.

We don't quite agree with the inference of Dr. Taylor's last statement, that this index is only applicable "to working men in the hot industries", because we feel the index has considerably broader application, for description of any conditions, indoor or outdoor, which result in heat stress.

As regards the last point mentioned by Professor Yaglou, whether we have made "premature claims," I think we will have to let our readers judge. This is a matter of opinion. We really only claim that the method yields a single value descriptive of heat stress, and that this figure takes into account the major factors which determine heat stress for lightly clothed men. We have already practiced the recommendation of Professor Yaglou that various available heat scales be compared. We went to Parris

Island, South Carolina, last summer to study the stress of heat on marine recruits, and we used 4 available indices in connection with that study.

We do not agree with Professor Yaglou's contention that this index is "an elaboration of a rational concept" originating from the Office of the Quartermaster General and designated the Thermal Acceptance Ratio. Using our nomenclature, TAR is $\frac{E_{max} - C - R}{0.75\ M}$; in words, TAR is the potentiality for removal of heat from the skin by the environment (represented by maximum evaporative capacity plus convective loss plus radiation loss) divided by the part of body heat production that is lost through the skin (75 percent). TAR when inverted does not resemble $\frac{M + R + C}{E_{max}}$, our HSI, because the expression 0.75 M is not the equivalent of $M + R + C$. The ratios yielded by TAR have no fixed meaning insofar as body heat stress is concerned unless M is a constant. For example, in different realistic situations, zero heat stress (thermal neutrality) may be represented by a TAR as low as 1.1 or as high as 8.0. In our opinion, TAR is in no sense a heat stress index.

In response to Mr. Hemeon, we agree wholeheartedly that the meaning of the index values in terms of strains on people requires further exploration. There are few data on heat tolerance limits for women, children, older people, or those with circulatory impairment.

In answer to Mr. Fleisher, I don't know of any work on the relationship between blood pH on one hand, and sweating or circulatory responses to heat on the other. We do not know that some people sweat more than others under apparently equivalent stress conditions. This appears to indicate a difference in efficiency of sweat for body cooling because those who sweat more apparently achieve no more cooling of the body than those who sweat at a lower rate.

I am just hazarding guesses here, but it may be that the sweat of a man with oily skin tends to remain in droplets rather than to spread uniformly over the skin, and that this means a smaller available surface for evaporation; in other words, such a man could not achieve as high a wetted area. He may also drip more, which would decrease the traction of the sweat available for evaporative cooling. Maybe hairiness also has something to do with the efficiency of sweating. It may be that water moves out along the hairs when it is secreted along the base of them. When water is evaporated at some point away from the skin, we know it isn't 100 percent efficient in removing heat from the skin; some of the heat of evaporation is removed from the surrounding air. These are possible explanations of apparent differences in efficiency of sweating. They are interesting things to look into; we certainly don't have the answers.

In answer to Dr. Hick, we think that there is a heat stress whenever there is a circumstance requiring compensatory action on the part of the body to prevent body temperature from rising. If a positive value derives from addition of metabolism radiation and convection, body temperature will go up unless the body resorts to protective measures. Any such situation therefore involves a heat stress.

In accordance with this definition, a man may be under heat stress without resulting harm. Actually on a warm day in Cincinnati, sitting at a desk in a non-airconditioned office, one may be subjected to a heat stress of anywhere from 25 to 35. The heat stress is real, but no injury normally results.

I think the question of Mr. Ash, regarding run-off has been partially answered. The index utilizes the relationship between evaporation required and maximum evaporation achievable from a completely wetted skin. As the required sweating approaches the amount that can be evaporated, more sweat will soak into the clothes and more will drip off, both evidences of loss of efficiency of the sweat that is produced. The results of this situation are a greater requirement for sweating and/or a rise in body temperature. The ratio reflects this relationship.

In conclusion, the authors wish to express their appreciation for the lively interest shown in this paper. Their future work along these lines will be guided by these thoughtful and useful comments.

1568

No. 1568

BODY EVAPORATION DURING SHORT EXPOSURES TO VARIOUS TEMPERATURES, HUMIDITIES, PRESSURES AND MASS VELOCITIES†

By J. W. McCUTCHAN*, LOS ANGELES, CALIF.

IT IS KNOWN that the human body's defense for high thermal stress is by increased perspiration and a limited amount of body temperature rise. This is quantitatively expressed by the first law of thermodynamics, which in symbolic form is: $Q - W = \Delta E$. The system is the human body (non-flow) and the boundary is the skin surface. Q is the heat transferred to the body; W, the work done by the body, and ΔE, the change in internal energy. In applying this equation to the human body it is customary to separate the various modes of heat transfer, to assume $W = O$ and to call ΔE the heat storage, q_s.
Thus,

$$\underset{\text{radiation}}{q_r} + \underset{\text{convection}}{q_c} + \underset{\text{metabolism}}{q_m} + \underset{\text{evaporation}^{**}}{q_e} = \underset{\text{storage}}{q_s}$$

Blockley et al[1] give data for human time tolerance at extreme temperatures as a function of body storage. The engineer can compute this storage by subtracting the heat loss by evaporation from the heat gained by radiation, convection and metabolism (a design value for pilots' metabolism is 75 kilocalories per (square meter) (hour). The calculation of the radiation and convection terms is done basically from the environmental variables, so it would be convenient if evaporation were likewise expressed in these terms. The object of these experiments was to derive an empirical expression for W_e, the skin water loss by evaporation, as a function of environmental temperature, humidity, pressure and mass velocity. The product of this skin water loss and the heat of vaporization will then give q_e.

It was decided to run a series of 50 experiments in which 5 levels of temperature, water vapor pressure, pressure (altitude), mass velocity and time would be controlled and the subject's weight loss by evaporation determined as the dependent variable. The range of variables was as great as practical with the equipment

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† This paper is the result of research sponsored by Wright Air Development Center (Aero Medical Laboratory) in cooperation with the University of California, Los Angeles.
* Department of Engineering, University of California.
** Note sign convention of thermodynamics which says heat transferred to the body is plus, thus q_e is a negative term.
[1] Exponent numerals refer to References.
Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

at hand. Oxygen was given to the subject for altitudes above 8000 ft to avoid hypoxia and to confine the low barometric pressure effects to those acting upon thermal exchange.

EXPERIMENTAL PLANS

The experimental schedule was based on 5 × 5 completely orthogonal squares making 25 experiments (Fig. 1). It was replicated once in order to determine the variance between subjects, making a total of 50 experiments. The independent variables of the environment were controlled at the specified settings. The principal dependent variable, weight loss, was measured by scale readings taken every

A	A	B	A	C	A	D	A	E	A
	A		B		C		D		E
A	A	B	B	C	C	D	D	E	E
A	B	B	B	C	B	D	B	E	B
	E		A		B		C		D
B	C	C	D	D	E	E	A	A	B
A	C	B	C	C	C	D	C	E	C
	D		E		A		B		C
C	E	D	A	E	B	A	C	B	D
A	D	B	D	C	D	D	D	E	D
	C		D		E		A		B
D	B	E	C	A	D	B	E	C	A
A	E	B	E	C	E	D	E	E	E
	B		C		D		E		A
E	D	A	E	B	A	C	B	D	C

	PRE PERIOD MIN.	TIME MIN.	TEMP °C	V P MM HG	PRESS MM HG	MASS KG/M² HR	VEL.	CODE
KEY	20	5	37.8	10	750	966		A
	15	10	48.9	15	636	1022		B
TEMP V P	10	15	60.0	20	522	1080		C
TIME	8	20	71.1	25	409	1140		D
PRESSURE V	6	25	82.2	30	295	1198		E

FIG. 1—ORTHOGONAL SQUARES EXPERIMENTAL DESIGN

5 min. In addition, the dependent variables of skin temperature, rectal temperature, heart rate and estimated water accumulation on the body were also measured and recorded.

Varying pre-periods as a function of temperature were used in the experiments. This was based on previous data which indicated this amount of time was necessary to reach a quasi-steady state at the respective temperatures. Actually a variable pre-period is inherent in the design because of the time lag in attaining the altitude setting after the chamber door is closed. Only in the 37.8 C and 48.9 C experiments was any time spent in waiting for this pre-period to elapse.

Another point of note was that all exposure times lasted for 25 min so that the time variable could have been rotated to other combinations than those specified in Fig. 1. However, since a quasi-steady state was postulated, the time variable had no significant effect upon the slope of the curves, and the experimental design permitted checking this fact.

The order of experiments was arranged to keep Wednesday open for equipment

repairs. Tests were run on Monday, Tuesday, Thursday and Friday, and no 2 high temperature experiments were run on successive days.

SUBJECTS

The 2 subjects (Table 1) were normally healthy male college students. Before each experiment, which was always scheduled in the morning, the night's sleep, diet and sedentary activity were checked as being usual and normal. Twenty min of sitting-rest at room temperature preceded the chamber exposures, but otherwise, the conditions were non-basal. Subjects were nude except for shorts, flight helmet and moccasins.

TABLE 1—SUBJECT CHARACTERISTICS

SUBJECT	SYMBOL	AGE (YRS)	HEIGHT (FT-IN.)	WEIGHT (LB)	SURFACE AREA (FT^2)
E. H.	plus	24	5′11″	181	23
F. R.	circle	23	5′7″	152	20.6

METHODS AND EQUIPMENT

Environmental Variables: The exposure chamber consisted of a one-man conditioned space with air temperature, air-movement, vapor pressure and ambient pressure control. The chamber is cylindrical in form, 70 in. long by 46 in. in diameter. The steel tank was lined with 6 in. of cork covered inside by galvanized iron painted with heat resistant, grey primer-surfacer. Equipment for evacuating the chamber to the equivalent of 40,000 ft altitude is located on the roof of the building.

Air movement through the chamber is controlled by a blower which can operate on an open circuit at sea level or on closed circuit when under vacuum. The air distribution system was designed to give reasonably uniform turbulence over the body. A series of tests was run in which air movements over the body were measured at a distance of 10 cm normal to the right shoulder, chest, lumbar, left elbow, right thigh, knee and left calf. The average of these readings was correlated with the flow-valve setting on the circulating blower, so that the mass flow was pre-set. This was done so that the subject had no duties during the run except to sit comfortably, thus giving more reliable scale readings.

Air temperature was controlled by a cycling type on-off controller, which is an electronic signaling device having a platinum resistance thermometer for a sensing element. Rapid recovery and inherent droop correction (an important factor after the door is opened) were the reasons for the choice of this particular equipment. Air and wall temperatures were sampled by parallel circuits of 5 distributed thermocouples and recorded by a potentiometer.

Humidity was controlled by an electronic dew-point hygrometer connected to a pulsating on-off steam valve. Wet- and dry-bulb thermometers were used as a check. Low humidity was maintained by passing a portion of the chamber air through a silica-gel dryer.

Physiological Variables: Evaporative loss was determined by weighing the subject sitting upon one *pan* of a double-beam balance. Electrical remote control of

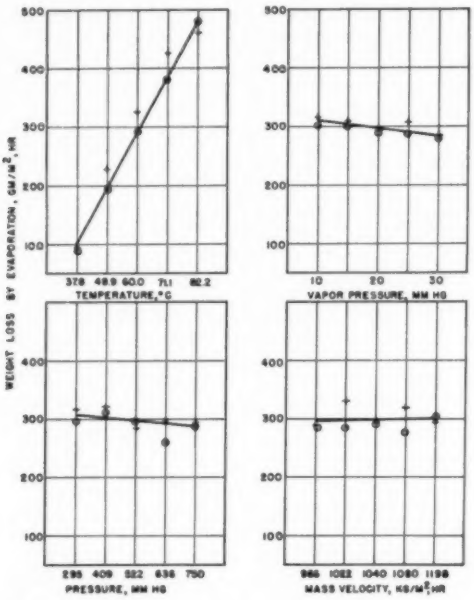

FIG. 2—VARIATION IN WEIGHT LOSS BY EVAPORATION (EACH POINT IS THE AVERAGE OF FIVE EXPERIMENTS)

balancing and subject stabilization between weighings is a unique feature of this balance system. The accuracy of the scales in simulated weight loss is ±5 grams. In order to measure both perspiration and evaporation it was necessary to measure the water which dripped from the subject and the water stored on the skin. Therefore, at the beginning of each experiment 2 towels were weighed, one was placed in the seat, and at the end of the experiment the subject was wiped off with the second towel. The gain in weight of the 2 towels in addition to that of the shorts and cap represented the difference between perspiration and evaporation.

With thermocouples held firmly against the skin, skin temperatures were obtained at the right scapula, lower abdomen, right forearm, left thigh and right calf. Recording was made with a potentiometer with an accuracy of ±0.25 C. The average of the 5 readings was designated the mean skin temperature. The rectal thermocouple was inserted to a depth of 10 cm and recorded on a precision potentiometer accurate to ±0.05 C. Heart rate was indicated on an electrocardiotachometer.

ANALYSIS OF RESULTS

The efficiency of this type of experimental design becomes apparent when the data are being reduced. To investigate the variation of weight loss with tempera-

ture, the data at each temperature level is used to obtain averages, which are plotted versus temperature. All other variables have their mean value, as for example 20 mm Hg humidity, 522 mm Hg pressure, and 1080 kg per(sqm)(hr) mass velocity. The data are then sorted according to the levels of the other variables and plotted. It should be pointed out that this procedure assumes that interactions between variables are non-significant. This is generally a satisfactory assumption, but if strong interactions do exist, they will be indicated by a large residual variation[2].

Average data points are given for the dependent variables of weight loss by evaporation, Fig. 2; skin temperature, Fig. 3; and rectal temperature, Fig. 4. Averages of the other variables are reported in Table 2. It was apparent from the weight loss data that in the 25-min test period the weight loss rate, even though tending to increase with time, did not increase significantly, so an average weight loss rate for the 25-min period was used in reducing the data.

Figs. 2, 3 and 4 show the variations as functions of ambient air and wall temperature (which were always the same), water vapor pressure, pressure, and mass

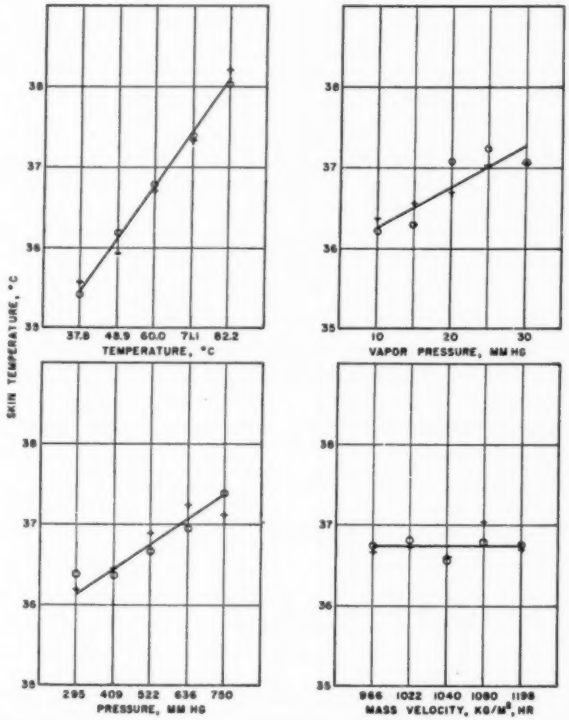

FIG. 3—MEAN SKIN TEMPERATURE (EACH POINT IS THE AVERAGE OF FIVE EXPERIMENTS)

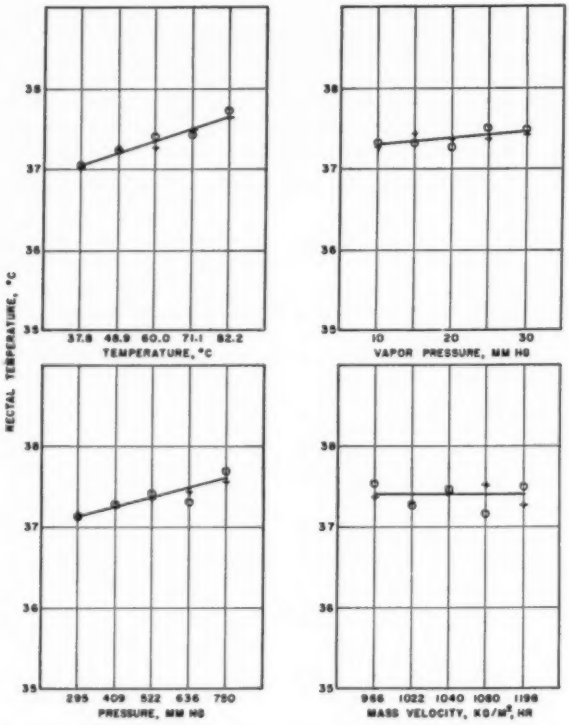

FIG. 4—MEAN RECTAL TEMPERATURE (EACH POINT IS THE AVERAGE OF FIVE EXPERIMENTS)

velocity. The lines are the *best fit* lines, and their slopes were taken to derive the following composite equations:

$$W_s = -150 + 8.70t - 2.09\,p_a - 0.079P + 0.0086G \quad (SE_{est}{}^* = \pm 35) \quad (1)$$

$$t_s = 30.51 + 0.0607t + 0.052\,p_a + 0.003P \quad (SE_{est} = \pm 0.30) \quad (2)$$

$$t_r = 35.71 + 0.0135t + 0.0096\,p_a + 0.0016P \quad (SE_{est} = \pm 0.25) \quad (3)$$

Since t_s and t_r are definitely functions of time, a time analysis was run and the slopes determined as $0.0149\,\theta$ for t_s and $0.0216\,\theta$ for t_r, but since these average slopes for 60 C do not predict accurately for the extremes of temperature, they were not included in the equations.

The relative importance of the environmental temperature variable is impressive. Table 2 gives the percentage variation of the dependent variables attributed

* SE_{est} is the standard error of estimate, and is the range which contains two-thirds of the data points.

to the respective independent variables. In interpreting this table the range of the independent variables as specified in Fig. 1 must be kept in mind.

If the range in vapor pressure and mass velocity had been greater, it is probable that these variables would have had a greater influence on the results, but these were based on the limitations of the equipment at hand, and at least the range in pressure is pertinent to environmental designs. If the design problem involves a greater range of water vapor pressure, attention is called to the work of Winslow et al[3], and a greater range in velocity was investigated by Nelson et al[4].

DISCUSSION OF RESULTS

It is customary in the analysis of biophysical data to draw analogies with physical systems and to express the results in similar form. The data from the present study were compared to the evaporation from single spheres, single cylinders and flat surfaces as summarized by Maisel and Sherwood[5]. Plots using the analogy of the evaporation from the body with a wetted cylinder[6] and a plot of the water accumulation on the skin against Colburn j_D showed great scatter of the points. It is believed, therefore, that the equation given in the previous section provides a much closer estimate of evaporation from the human body in the specified range.

A question which still remains unanswered is: Why does the man subjected to heat stress appear so much better off at low ambient pressure? It has been suggested[7] that the evaporation rate increases, which in turn stimulates greater skin water loss which should result in greater weight loss due to evaporation as altitude increases. These data do not bear this out, at least not for a 25-min. exposure preceded by a preperiod of approximately 10 min. A suggested explanation is that it is the diffusion loss that is increased at low barometric pressure. This is without time lag so that the body temperature rise is slower and the rate of increase in sweat gland activity is diminished with the result that the body supplies about the same amount of water at an overall lower body temperature.

CONCLUSIONS

1. Over the ranges tested, with 2 healthy young men as subjects, environmental temperature has by far the most significant effect on skin water loss by evaporation.
2. The beneficial effect of reduced pressure on body temperature and hence the rate of heat storage is also significant.
3. Here is an example of an economical method of designing a physiological experiment, using the *Latin Squares*.

TABLE 2—PERCENTAGE VARIATION

AVERAGE VALUES OF	W_e	W_s	W_p	t_s	t_r	H. R.[a]
Temperature	97.4	65.7	97.0	75.0	56.3	89.4
Subject to Subject	0.5	2.2	0.5	0.1	5.0	2.3
Pressure	1.1	7.8	0.4	12.3	29.0	3.6
Vapor Pressure	0.2	18.5	1.2	9.3	2.7	1.8
Mass Velocity	0.5	2.2	0.4	1.7	5.2	1.6
Residual	0.3	3.6	0.5	1.6	1.8	1.3

[a] H.R. is heart rate in beats per minute.

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NOMENCLATURE

D = characteristic dimension, meters.
D_m = diffusivity, square meters per hour.
G = mass velocity ($G = V\rho$), kilograms per (square meter) (hour).
K = vapor conductance, grams per (square meter) (hour) millimeters Hg.
p_a = vapor pressure of atmosphere, millimeters Hg.
P = barometric pressure, millimeters Hg.
R_e = Reynold's number (DG/μ), dimensionless.
t = air temperature, degrees centigrade.
t_r = rectal temperature, degrees centigrade.
t_s = mean skin temperature degrees centigrade.
V = velocity, meters per hour.
W_a = water accumulation on the skin, grams per (square meter) (hour).
W_e = evaporative water loss, grams per (square meter) (hour).
W_p = perspiration loss ($W_p - W_e + W_a$) grams per (square meter) (hour).
ρ = density, grams per cubic meter.
μ = viscosity, grams per (meter) (hour).
θ = time, minutes.
j_D = $(KP/G)\ (\mu/\rho\ D_m)^{0.67}$

DISCUSSION

DR. DAVID MINARD†, Bethesda, Md., (WRITTEN): 1. The author represents an institution and group which has been in the forefront of studies on human tolerance to high levels of environmental heat stress.

2. The excellent control of the experimental conditions and the excellent design of the experiments are both reflected in the consistency of the data and the ease with which the data are handled.

3. It would appear that the tolerance time of the most severe conditions (82.2 C) established the time period for all experiments (25 min.). This precluded the establishing of a steady state of the body under the various environmental states. The author recognizes this and has attempted to estimate rate of change in the physiologic variable at the end of the tests. However, this limits the usefulness of the data in estimating

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the various quantities of heat lost by evaporation and by radiation and convection, as opposed to that being stored. It would have been of interest, though not essential, to the requirement of the experiment, to know what the metabolic load was, and to know whether it varied under varying conditions of pressure, temperature, humidity and air movement. For comparison with similar data of others, resting levels under pre-test conditions for each physiologic variable would have been helpful. Plotting heart rate against temperature, vapor pressure, ambient pressure, and air movements also would be of interest to physiologists.

4. The effectiveness of air movement and humidity in body heat exchange and in affecting evaporation rate is well documented. The relative ineffectiveness of these factors in influencing evaporation rate in these tests can be attributed to the narrow ranges of vapor pressures and mass velocities studied. It does not seem surprising, therefore, that the composite equations for relating temperature, humidity, and air movements on evaporation, skin, and rectal temperature point to the predominant role of temperature.

5. It is of interest to observe that evaporation rate changes little with barometric pressure whereas both skin and rectal temperature decline as barometric pressure falls. The author suggests that loss of moisture by diffusion is increased as opposed to sweat gland activity which is diminished. This reviewer is not clear in his mind how the author distinguishes in these tests between the amount of moisture lost by diffusion (which the reviewer assumes to be the same as insensible perspiration) and sweating.

6. It would be of interest to know the levels of heat stress which these various conditions represent. This might be done utilizing the recently published method of Belding and Hatch*. Since the present author measures both total sweating and evaporation his data might also serve as an added check of the applicability of this new method of Belding and Hatch for evaluating environmental heat stress.

THEODORE HATCH‡, Pittsburgh, Penna., (WRITTEN): The failure of the evaporative weight loss to vary, in the reported experiments, in proportion to the skin-air vapor pressure difference is a significant point in considering the general applicability of the proposed equation relating weight loss to the several physical parameters of the environment. This occurred despite the very considerable heat load on the subject which, in all but the coolest of the test conditions, called for maximum sweating and evaporation. Using the Fort Knox heat transfer coefficients for man (reference 4), one finds for the middle temperature and velocity (60 C, 50 fpm) for example, that the necessary skin temperatures to balance $M + R + C$ against the maximum evaporative capacity are: 37.8, 40.3, and 42 C for 10, 20, and 30 mm vapor pressure, respectively, well above the reported skin temperatures after 25 min. exposure.

Thus, it appears that during the relatively brief period of early exposure with the system in an unsteady state, the output of sweat did not keep up with demand. The heat deficit was made up by body heat storage. Thus, the evaporative rate was fixed by the sweat output and could not vary appreciably, therefore, with vapor pressure difference. It would be interesting to know how the evaporation rate at the end of the exposure period compared with the initial rate. Was there a lag, in fact, in sweat output relative to demand with the sweat rate fixed, perhaps, by the degree of rise in body and skin temperature, rather than by the thermal demand of the environment? I believe that such a lag has been shown in previous studies. The question is of great practical importance in considering the thermal behavior and welfare of industrial workers exposed intermittently and for brief periods to high environmental temperatures.

In view of these comments, the proposed equation appears to have only limited application. It describes the average initial sweating response to heat load over a 25 min

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* H. S. Belding and T. F. Hatch, "Index for evaluating heat stress in terms of resulting physiological strains", JOURNAL SECTION, *Heating, Piping and Air Conditioning*, Aug. 1955.

‡ University of Pittsburgh and Mellon Institute.

period, with the body in an unsteady thermal state but does not express the basic capacity of man to exchange heat by evaporation with his atmospheric environment. The latter was not the objective, of course, of the study.

A word of caution should be voiced concerning the use of the Latin Square technic in experimental design. Two important requirements must be met. First, this procedure requires that each of the variables operate according to a continuous relationship over the experimental range. Obviously, discontinuity in the relationship cannot be tolerated. Consider the influence of vapor pressure, for example, upon the thermal behavior of man. For a given level of activity and clothing, the vapor pressure is not a limiting factor so long as the environmental temperature and velocity are below certain levels, whereas, above these levels, it is the controlling factor. Clearly, the experimental range must not span this dividing line if the Latin Square technic is to be used. The technic also has questionable application in studies on man in an unsteady thermal state, for the same reason of a changing relationship as the experimental factors are varied. Second, the Latin Square procedure requires a linear relationship between each of the variables and the measured response. To illustrate: one cannot expect to demonstrate the well-established fractional power relationship between air velocity and convective heat exchange from a Latin Square analysis of heat exchange over an arithmetic range of velocities. A similar difficulty arises with respect to analysis of environmental temperature and vapor pressure. Heat exchange rates bear linear relationships with the skin-air differences in these factors and changes in their values at the skin are quite as important as the experimental changes in the environment. As the latter vary from one experiment to another, there will not necessarily be a directly proportional change in magnitude of response. The error may be quite great when the skin-air difference is low. The air vapor pressure along Eichna's limiting tolerance line*, for example, does not vary with air temperature in the manner predicted by Haldane (fixed by wet bulb temperature) for the reason that, as air temperature increased, the skin temperature also increased and, in consequence, an increasingly higher air vapor pressure was permitted with increasing air temperature to balance the rate of evaporative cooling against $M + R + C$.

Such findings would not be brought out by the Latin Square technic unless they were postulated in advance and the experimental design set up to test for them. As the author points out, there is great economy in the use of this procedure but the limitations in its application must be kept in mind.

DR. L. P. HERRINGTON**, New Haven, Connecticut, (WRITTEN): All research groups concerned with the physiological and biophysical problems of thermal stress have been conscious of the industrial, civil aviation, and military demand for heat loss data on human beings at altitude. Such data is needed both for tolerable conditions and for high stress exposure of limited time tolerance. This paper provides a valuable set of data for heat stress conditions ranging from the upper margin of long term tolerance to about 80 C., with simulated altitudes ranging from ground level to a pressure level of 295 mm Hg.

For all practical applications the safeguards provided by the statistical design of the experiments and the internal consistency of the data make the reported data immediately available and useful for aviation air conditioning engineers, and others concerned with thermal stress at variable altitude. For more remote but none the less important research problems, a supplementary publication by the author giving the mean wall temperatures and air velocity measurements would be widely appreciated. This would permit the analytic connection of the data of this excellent study with the numerous reports from ASHAE., the Russell Sage Foundation, the John B. Pierce Foundation, and military groups, in which the partition of heat exchange has been related to radiation and convection terms separately, utilizing a velocity factor.

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* Eichna, L. W., et al: The upper limits of environmental heat and humidity tolerated by acclimatized men, working in hot environments. *Jour. Ind. Hyg. and Toxicol.*, 27, 59, (1945).

** Director of Research, John B. Pierce Foundation.

One of the most interesting points of this paper is the considerably lower body and skin temperatures for comparable air temperature stress as the simulated altitude increases. If we use a weighing factor of 0.80 for rectal temperature, and 0.20 for skin temperature, the body mass temperature appears to be approximately 0.7 C deg lower at the extreme simulated altitude as compared with ground level. A specific heat for the total body of 0.83 would thus indicate that at extreme altitude the reduction in heat stress for comparable air temperature is of the order of 75.5 kg. x 0.7 C x 0.83 or approximately 44 kg cal in an exposure period of 25 min.

The total heat production of the subjects is of the order of 75 kg cal per hr and the additional cooling at the extreme altitude appears to be at a rate of approximately 105 kg cal per hr. The observed evaporation increase of 10 to 20 grams per(sq m)(hr), as the author notes, cannot account for additional cooling of the order found.

This unexplained effect does not in any way reduce the importance and immediate usefulness of this valuable new data obtained with excellent apparatus and informed experimental design. It does invite, however, determination by the author or others, of the source of the extra cooling, particularly so since the author reports that the subjects were less stressed, or felt better, at the higher simulated altitudes, presumably under comparable temperature exposure. A very approximate conversion of the mean mass velocity to an estimate of linear air velocity indicates that the mean cross sectional air velocity was low. Under such conditions the coefficient of heat exchange by radiation is of the same order or larger than the body's convection coefficient. This factor might change the operative temperature index of the experiment, and in the event of cooler walls, at altitude, might explain this interesting result.

AUTHOR'S CLOSURE: The author appreciates the clarifying comments and the questions raised by Dr. David Minard, Mr. Theodore Hatch and Dr. L. P. Herrington. Dr. Minard noted that the time of exposure was established by the most severe condition, which was true. We were comparing 100 F environments with 180 F environments and felt that equal exposure times provided a basis for this comparison.

Metabolic rate measurement would have been desirable except that we considered the scale readings on body weight loss more important. This precluded the use of hoses and the accessory equipment necessary to measure the metabolic rate. Furthermore, we felt this could be estimated [50 to 60 Kcal per(sq m)(hr)] from the studies of R. Jones and C. L. Taylor which were published as a Wright Air Development Center Technical Report, July 1955.

Heart rate was actually measured and the data summarized in the following equation:

$$\text{H.R.} = 23 + 0.83\,t + 0.22\,p_a + 0.017P + 0.004\,G \quad (\text{SE}_{est} = \pm 7)$$

Dr. Minard asked how we differentiated between skin water loss by diffusion and sweat gland activity. We haven't as yet. This will be the basis of a future investigation. At present, this explanation is just a theory, based on the work of G. E. Burch and T. Winsor; also E. A. Pinson; both studies measured skin water loss with sweat glands inactivated.

Mr. Hatch mentioned the lag in the onset of sweating when a man is exposed to thermal stress. Certainly we had this lag. However, due to placing the man in the chamber and attaining the proper altitude and temperature settings, we were unable to take reliable weight readings during this period. After the environmental variables stabilized, the linearity of the weight loss data was surprising.

It should be noted that linearity is not an inherent limitation of the Latin Squares technique. D. N. Harris, F. R. Watson and T. Frame-Thompson in their report on Preplanned Tests With Magic Squares demonstrate the consideration of higher power terms.

Dr. Herrington commented on the wall temperatures. We attempt to control air and wall temperatures at the same value. This is done by *soaking* the chamber about 10 deg above the experimental setting. After the chamber door is opened and the subject enters we find that this technique allows us to attain the desired temperature

settings in a minimum time. A run is considered satisfactory if the air and wall temperature are held within ±2 F deg of each other.

The question of the use of air mass velocity, G, instead of linear velocity, V, was raised. The reason we used G instead of V was because convection heat transfer correlates better with mass velocity for the pressure range used in these experiments. $G = V\rho$ so that the linear velocity can be calculated for any experiment by dividing by the air density, ρ.

1569

No. 1569

HUMIDITY EFFECTS ON THE ODOR PROBLEM

By RICHARD L. KUEHNER*, YORK, PA.

THAT HUMIDITY affects the sense of smell or the production and maintenance of odor levels has long been a conversation piece in the air-conditioning industry. The hound dog that works effectively only on humid nights and smells bad only when wet, or the air-conditioning installation which develops offensive linoleum odors under high moisture, but not under dry conditions, suggests that an approach, if not the solution itself, to the odor problem might be found in proper humidity control. The ASHAE Laboratory is now making a fundamental approach to this problem. What limited previous work has been done is of a nature not designed for useful engineering generalizations.[1,2,3,4,5]

The research discussed here was undertaken to explain certain inconsistencies in odor measurement found during the practical evaluation of potential deodorants. It was designed to gain sufficient knowledge of the control of experimental conditions to yield reproducible answers in formal odor studies. However, the results indicate certain facts of general and immediate use.

Existence of an odor problem depends primarily upon 3 basic factors: (*a*) the rate of odor production; (*b*) the natural escape of odors from enclosed spaces (as opposed to purposeful removal of odor by deodorization or ventilation); and (*c*) the olfactory sensitivity of subjects perceiving the odor. Relative humidity exerts a major effect on each.

EXPERIMENTAL PROCEDURE

Escape of Odor: Test work was undertaken at York, Pa. in a 1700 cu ft room of standard construction. There was one outside, 2-course brick wall of normal studding, wood lath and plaster, and 2 windows; 3 inside walls with plaster and wood lath on both sides of the partitions, and 3 doors. Inside temperature was automatically maintained at 80 F by a combination of steam coils and electric heaters. Relative humidity was maintained at 25 percent and 50 percent by boiling distilled water in the space and condensing the excess moisture with standard humidistatically controlled refrigeration equipment. No ventilating air was used. Each combination of inside wet- and dry-bulb temperature was termed a test. Outside temperatures and humidities were uncontrolled. Tests were made only on days when desired outside wet and dry bulbs existed and only when there was insufficient air motion to alter normal air infiltration appreciably. After setting

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[1] Exponent numerals refer to References.

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conditions, cigarettes were mechanically burned at the rate of 2 per hr (2 g per hr) starting at zero time. This rate simulates normal four-man smoker occupancy. Odor measurements were taken at intervals of 4 and 7 hr from zero time.

Olfactory Sensitivity: Variations in the sensitivity of the nose were determined at the same time and using the same test room just described (chemical measurement of odor was taken periodically to establish that odor had actually left the space under certain test conditions). At each selected odor level, 2 measurements were taken; one, using dilution air in its normal, dry purified state; the other, using dilution air humidified to 25, 40 or 50 percent RH (relative humidity). Dilution air was conditioned by passing purified air through heated distilled water, condensing to the proper dew point and reheating to 80 F in water baths. Dilution air and room air were always maintained at 80 F; room air always at 50 percent RH. A test consisted of the measurement of a given odor level in the test room by dilution air at zero percent RH and also at one other relative humidity.

To get a rough extension of the plot of this data, additional odor measurements were made by operators who, inhaling through the nose, had breathed air at zero relative humidity and air saturated with steam in excess of 100 percent RH, immediately prior to sampling. The latter was accomplished by breathing over beakers of boiling triple-distilled water, making certain that the air breathed was not appreciably above 80 F.

Odor Production: The objective was to establish whether or not standard odorous household materials impart different odor concentrations to the air under different relative humidity conditions. Tests were run on 1000 g of onions, 2000 g of carpeting and 1200 g of linoleum, each contained separately in a non-coated steel drum of 2.65 cu ft. After placing the test material in the drum, air, conditioned to the desired experimental temperature and humidity, was flowed through at a rate of 0.58 cfm for 4 hr. This permitted the contained test material to adjust to the experimental conditions. Ventilation was then stopped, and the drum and contents isolated by valves. For each series of tests, odor measurements were taken at 3 hr and 20 hr from this point. Ventilation air and drum were always maintained at 80 F; zero percent and 50 percent RH ventilation air was used.

Odor Measurement: Odor measurements were taken by the dilution-to-threshold method, using techniques previously described[6]. This consists, briefly, of drawing odorous air by blower from the test room or chamber, diluting this odorous air with measured volumes of purified dilution air until the odor is just detectable by the operator. Dilution air was purified by the activated carbon, dry ice technique[2]. This dries the air to less than 4 grains of moisture per lb of dry air and for convenience is referred to here as zero RH. It has been previously noted that the dilution system is open to question[6,7] in that an odor measurement is valid only if the level of sensitivity of the operator is known and that this sensitivity does not vary from test to test. In this investigation, to insure against such experimental errors, measurements were always taken by 2 or more operators whose level of sensitivity was established by one or more of the following techniques. Two operators of known and similar sensitivity checked each other for a given odor level, or each was demonstrated to have a uniform sensitivity to xylene or tobacco odor before each test. These checks were previously described[6].

All odor measurements, excepting those demonstrating the effect of humidity on the sense of smell, were made with purified dry dilution air. The sensitivity of the operators used was: 0.001 g of tobacco burned per cu ft of air measured one threshold of odor; air saturated with xylene at 32 F measured 380 thresholds.

EXPERIMENTAL RESULTS

Escape of Odor: As published earlier[6] it was found that with a standard smoking rate of 2 cigarettes per hr, the odor level in a test space would reach 14 thresholds at the end of 9 hr. This work was done in the summer. Subsequent tests run in winter indicated that the odor level would not greatly exceed 1 threshold with 2–4 times this smoking rate. Infiltration and exfiltration as commonly thought of were readily eliminated as causes. Accordingly, examination of humidity difference between inside and outside conditions was made as a formal study. Table 1 summarizes the data. From this the water vapor pressure of each location was calculated. The table compares the odor level differences at various inside-outside water vapor differences. Since the smoking rate in all tests was identical, the effect of this humidity differential becomes apparent. The highest thresholds were found when water vapor pressure outside was equal to or exceeded that of the inside. With each increase of inside vapor pressure over outside there was a concomitant loss of odor. Tests Nos. 1 and 10, which approximate winter and summer operating conditions, illustrate the divergent odor behavior in these two seasons.

Olfactory Sensitivity: In a pure approach the sensitivity of the nose at zero RH should be compared with that at 100 percent RH. Since this was a practical test with the odor room (*i.e.* odor sample) always held at 50 percent RH, the achieving of zero RH air to the nose was impossible no matter how much dried purified air was used to dilute the odorous air. Further, the mixture of dilution-odorous air required to measure 10 thresholds is always drier than the mixture used in measuring 2 thresholds. It becomes apparent that the so-called dry air measurement is actually a flexible baseline depending upon the odor level to be measured. A series of laboratory tests established that variations in water vapor between 4 and 30 grains per lb of dry air had little effect on olfactory sensitivity. This meant, that for purposes of this investigation, odor levels of less than 3 thresholds could not be used. Less than 2 parts of dry air per part of odorous air at 50 percent RH is insufficient to reduce the grains per lb to less than 30.

Table 2 contains selected data showing the general experimental conditions and the range of variation. It demonstrates the method of arriving at percentage reduction of olfactory sensitivity. Odorous air is the odor sample drawn from the

TABLE 1—VARIATION IN ODOR LEVEL WITH WATER VAPOR PRESSURE DIFFERENCE

Test No.	Inside Conditions			Outside Conditions			Odor Level		$V.P._1$-$V.P._2$
	Temp. F deg	RH %	$V.P._1$ mm Hg	Temp. F deg	RH %	$V.P._2$ mm Hg	At 4 hr	At 7 hr	
1	80	49	12.80	29	41	1.70	+1	+1	11.10
2	80	50	13.05	40	42	2.64	+1	+1	10.41
3	80	50	13.05	36	53	2.85	+1	+1	10.20
4	80	50	13.05	40	46	2.89	+1	+1	10.16
5	80	50	13.05	58	47	5.78	2.4	3.3	7.27
6	80	50	13.05	62	44	6.23	4.3	6.1	6.82
7	80	23	6.01	34	41	2.05	6.2	6.7	3.96
8	80	27	7.05	38	78	4.56	5.8	7.3	2.49
9	80	30	7.84	55	68	7.54	6.7	13.6	0.30
10	79	50	12.75	75	60	13.27	8.4	14.0	−0.52
11	80	50	13.05	85	52	15.99	8.8	15.9	−2.94

TABLE 2—EFFECT OF HUMIDITY ON ODOR MEASUREMENT

ODOROUS AIR		WITH DILUTION AIR A				WITH DILUTION AIR B				% DECREASE IN SENSITIVITY
TEMP. F DEG	GRAINS WATER	TEMP. F DEG	GRAINS WATER	GRAINS IN MIXTURE	THRESHOLD	TEMP. F DEG	GRAINS WATER	GRAINS IN MIXTURE	THRESHOLD	
80	78	80	3	20	4.3	80	78	78	2.5	42
81	77	80	3	15	6.1	80	78	78	3.4	44
79	75	79	4	19	4.7	79	58	63	3.8	19
82	81	81	3	13	8.1	82	65	67	6.8	16
79	74	81	3	11	8.4	82	40	45	7.4	12
80	78	80	2	11	8.8	80	37	41	9.3	+6

standard condition test room. Dilution air A column records the odor level measured with dry dilution air, and dilution air B column records measurement of the same odor level with humidified dilution air. The final column is the percent depression of the sense of smell by increased water vapor content. The data are also grouped vertically to show this depression when the dilution air was humidified to 50, 40 and 25 percent. In all cases the grains of moisture to the nose in the dry dilution air measurement was well below the critical point of 30, so all of the comparisons can be considered valid.

Fig. 1 is a plot of the total data, each point comprising an average of as many as 20 determinations. It illustrates the increasing diminution in the acuity of the sense of smell with increasing water vapor. The dotted extension shows only that sensitivity to normal odor levels does go to extinction in extreme conditions. Because the upper point was determined by a different experimental procedure, it does not strictly compare with the other points on the curve, hence the dotted line.

Odor Production: The final series of tests involved holding various odorous materials in closed vessels at low and at high relative humidities. Linoleum gave the most outstanding results. Table 3 records the odor level reached with similar quantities of linoleum held at zero and 50 percent RH for 3 hr and 20 hr. Each odor level is an average of several determinations. The results are apparent.

This conclusion could not be so factually substantiated by tests on carpeting and onions. With carpeting, no odor appeared under dry air storage. At 50 percent RH, one threshold of odor developed in 24 hr storage. With onions, inconsistent results were found though odor levels were generally higher at high humidity than at low. It was later found that the onions themselves were imparting sufficient humidity to the sample chamber to upset the experimental conditions. An interesting side light of the onion tests should be mentioned. Following each test the sample chambers were ventilated with dry purified air for 24 hr to remove adsorbed odor. It was found that following this treatment so

TABLE 3—THE EFFECT OF HUMIDITY ON THE VOLATILIZATION OF LINOLEUM ODORS

TIME HOURS	ODOR LEVEL AT	
	0% RH	50% RH
3	10.6	25
20	17.9	27

long as the sample chamber was kept dry it remained odorless. When the humidity in the vessel was increased to 50 percent the odor of onions appeared. This could be repeated several times.

CONCLUSIONS

High humidity reduces the acuity of the sense of smell and accelerates the volatilization of odors from certain household substances. Increasing the water vapor differential between the air of an enclosed space and that of a surrounding outside space increases the rate of odor loss from the inside space. This imposes previously unrecognized requirements upon the odor researcher. It also gives immediately useful information to the application engineer in understanding certain odor phenomena and suggests potential solutions to specific odor problems or methods of improvement for general odor problems.

It becomes essential that the researcher record wet- and dry-bulb temperature of both inside and outside air to make his work of value to others. For example, Yaglou's recent work[8] showed ventilation requirements for controlling tobacco

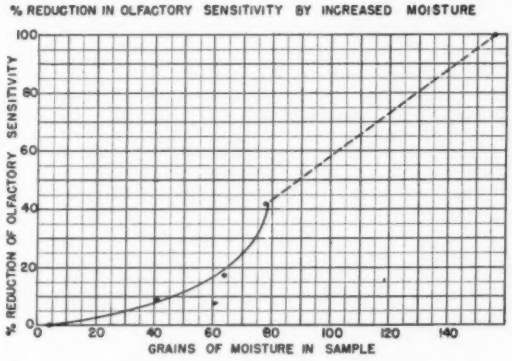

FIG. 1—PERCENT REDUTION IN OLFACTORY SENSITIVITY BY INCREASED MOISTURE

smoking odor roughly 50 percent greater than previous findings[6]. If the vapor pressure differential of inside over outside were greater in the earlier report than in the latter, a satisfactory explanation is found.

Additionally, use of properly dried dilution air is required for dilution-threshold measurement of odors. Most commercial and experimental dilution apparatus provide methods for deodorizing the dilution air but make no allowance for drying it. This means that in using atmospheric air for dilution, the same concentration of an odor is measured lower on a high humidity day than on a low one. The data here presented indicate that fairly consistent odor levels are measured only when the moisture content of the air to the nose is less than 30 grains per lb of dry air. Above this level slight changes in humidity appear to give major changes in olfactory acuity.

For experimental work on the odor generating properties of materials, a careful control of humidity is necessary, since the rate of odor volatilization varies with

humidity. This phenomenon has useful functions. Where background odor arising from the intrinsic odorous qualities of the construction materials of a test space becomes a problem, a reduction in the ambient relative humidity reduces the rate of odor production. Conversely, where odor absorbed on surfaces complicates an experimental program, ventilating the space with high humidity air will help cleanse the surfaces.

For practical odor problems three humidity effects, one negative and two positive, have been demonstrated. The question is, which direction gives the most benefit? Is the olfactory depressing effect of high humidity more beneficial than the opposing odor generating effect, or the reverse? Where the odor generating source is independent of water vapor effect, such as smoking and cooking, a high relative humidity is a consistent advantage in depressing the sense of smell and increasing the vapor pressure differential effect on the loss of odors. Where the odor source is intrinsic, as is the case with linoleum, paint and upholstering, reduction of relative humidity would be beneficial.

Earlier knowledge specified 80 F summer and 70 F winter conditions, but current thinking is shifting toward a 75 F year-round temperature for human comfort requirements. It is equally conceivable that the standard of 50 percent RH may not be the most desirable comfort condition from the odor standpoint.

This work was undertaken for a specific purpose—to better control odor studies. However, the practical signposts are here, but they are only signposts. Until a curve can be drawn showing percent change of olfactory sensitivity with small changes in humidity, and the precise degreee of odor loss or generation on many more practical substances defined, the individual engineer must use his own judgment in the application of this knowledge to each odor problem. The data here presented permit the exercising of a judgment heretofore unrecognized.

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8. Ventilation Requirements for Cigarette Smoke, by C. P. Yaglou (ASHAE TRANSACTIONS, Vol. 61, 1955, p. 25).

DISCUSSION

W. L. FLEISHER, New York: In the odor problem I wanted to know whether the same person tried to evaluate the odor. In other words, in my early days I was asked to

check an odor discharge at a rendering plant. When I went into the plant I nearly died. I sought the superintendent and said, "How do you live in odor like this." He said, "What odor?".

I was sent down to measure odor in a chocolate plant. In fact it was Hersheys before they moved and I, for two weeks afterwards, could smell nothing but chocolate and I feel that unless a different person tried to evaluate the difference over a period of time they wouldn't have any idea whether it was a change in odor or not. In other words, it would have to be done by 2 different people or you would have to have different people to evaluate the odor at different times, because odor is most persistent and one person judging odor over a different period of time wouldn't be capable of recognizing any change.

The measurement of odor by olfactory means is a very difficult problem and certainly can't be evaluated by one person.

C. S. LEOPOLD, Philadelphia, Penna.: Dr. Kuehner, if you repeated these experiments would you deliberately select a leaky room of this small size or would you take precautions to prevent the large infiltration which apparently occurred during your tests?

T. H. URDAHL, Washington, D. C.: As chairman of the Technical Advisory Committee on Odors, I first want to express our appreciation of Dr. Kuehner's paper and we would most appreciate any papers we might receive as a result of research from whatever basis that would throw a little more light upon various approaches to a problem that can at best be described as nebulous, though popular.

Having been associated with this technical advisory committee on odors for some time, and since odors apparently ring a bell in people's memories, I have even been introduced as chairman of the Royal Order of Skunks, because the gentleman forgot the Technical Advisory Committee on Odors of American Society of Heating and Air-Conditioning Engineers and I was so introduced at a Rotary meeting in California, and then asked to take over and explain what it was all about. I have one observation to make on what Dr. Kuehner has pointed out as a result of some of his work. It resulted again from my being chairman of the committee. They think, well, this fellow should be able to write a prescription on odors. He knows something about it.

I think because I know nothing about it, that's why I took on the job. Nobody seemed to want it. Sometime ago, my very good friend, Mr. Reynolds, formerly Commissioner of Public Building, General Service Administration called me and said he had a very dear friend who had a very and extremely annoying odor problem and could I possibly look at this problem and do something that might alleviate it. Well, as usual these things come just as I am about to leave and had just an hour or so to look at this problem which consisted of a residence, very well built, very tightly constructed, and everything very well weather-stripped; not air-conditioned but heated with a warm-air heating-system.

The owners were unable to live in the residence. They had taken a suite at the Mayflower Hotel. The general characteristics of the odor one noticed on coming in was that of fresh horse manure. The composition, contributing composition of this odor I found by using my nose, was a fresh coat of paint throughout the house. The house had just been redecorated with one of our new paints. They also had laid a new vinyl floor in the kitchen. The vinyl tile gave off an odor rather characteristic of an operating room that had become very stale. That, mixed with the odor from the paint, resulted in a composite odor as I said, resembling fresh horse manure.

The owner of the house, however, complained that he was allergic to the gases that came out of whatever it was. His eyes watered and he just couldn't live in it. The other people in the family just complained because they didn't like the smell. Now the prescription that I picked out of the air was to tell these people to keep out of the house during the day; set the thermostat at 100 F. If they could get it as hot as they could they might evaporate a lot of water into the space. Didn't have any humidifier but kept a kettle on the stove; kept the house tightly closed until dinner time and then came home and opened the windows about 4 inches; turn on the attic fans; set the thermostat;

go out for at least a 3 hour dinner and come back and if they could stand it, sleep. I heard later that it worked very well. They repeated it twice and the odor disappeared.

This prescription I practically fished out of the air and in a sort of a way it bears out the things that Dr. Kuehner observed. In extracting these odors from substances such as some of these new vinyls, paints, and of course pulling them out in the air and extracting the air, one can get rid of them.

AUTHOR'S CLOSURE: Mr. Fleisher is quite correct in his comments suggesting that odor levels evaluated by the unaided noses of several persons are not necessarily comparable or even of comparative value. Further, because of fluctuation in olfactory sensitivity, one person's odor perceptiveness is not necessarily the same from day to day or even hour to hour. In this work the nose was used because it is the most sensitive odor-detecting device available; the dilution to threshold technique was used because this permits the use of numerical values, rather than adjectives, in recording odor concentrations. However, for the reasons stated dilution to threshold alone is not sufficient, the sensitivity of the evaluating nose must be defined or standardized. This can be done by a method reported in my earlier work (ASHAE Research Project No. 1466-ASHAE TRANSACTIONS, Vol. 59, 1953, p. 77, and Annals of the New York Academy of Sciences, Vol. 58, Article 2:175–186, March 24, 1954). Because of this standardizing technique the odor levels recorded in this paper are valid and the results can be duplicated by anyone whose odor sensitivity has been established or mathematically corrected.

If we were setting up a test program to establish individually the various humidity effects revealed by this work, we would not "deliberately select a leaky room of this small size" for all phases. As pointed out this is one of the rooms that we use continuously for deodorant testing; throughout the year variation in odor behavior was found. As revealed in this paper it was the various humidity effects that were altering standard odor behavior. The studies were performed in this leaky room because it was in this room we wanted to make technical use of the information. However the room is considered to be no more leaky than the average room and it is only because it was of standard construction that the conclusions could be drawn concerning the diffusive loss of odor from standard occupied spaces. This item could not have been revealed in a vapor-tight room. The size of the room should have no effect since final conclusions were drawn on odor levels rather than absolute quantity of tobacco burned.

Mr. Urdahl's observations do indeed substantiate the findings recorded in the paper wherein high humidity accelerates the volatilization of odor from intrinsically odorous substances and the release of secondarily absorbed odors from surfaces. One comment I might add is that I would be interested in anyone's observations either now or by correspondence dealing with experiences or observations on any relationship between humidity and odor.

There were 3 ultimate objectives in publishing the information contained in this paper; all designed for furthering investigations of odor problems. First, is to help clarify the common belief that odor because it involves a physiological response is a nebulous problem. It is felt that this problem is not nebulous. It is merely that we are ignorant of the variables which give the impression of intangibility and uncontrollability. This paper reveals some of these variables and their effect. Secondly, the dilution to threshold technique to us has always been the simplest and cheapest method of odor measurement and one yielding numerical values unrelated to any psychological interpretation. Years ago this technique was popular but fell into disrepute because of variability in olfactory sensitivity. The results of this work suggest that possibly it isn't unharnessable physiological sensitivity which is varying but that perfectly controllable factors such as humidity are influencing odor concentration and nasal acuity. It is suggested that those who have tried the dilution to threshold technique and found it wanting repeat their tests with proper consideration of humidity. Finally, the influences described were determined not only in the laboratory but have been observed in field installations. The suggestion is made that those of you who have specific odor problems turn your attention to humidity as a possible solution.

1570

No. 1570

ELEMENTS OF DUAL-DUCT DESIGN AND PERFORMANCE

By N. S. SHATALOFF*, NEW YORK, N. Y.

THE MOST important function of the dual-duct system is to provide flexible temperature control to a multiplicity of zones and spaces fed by one or more systems. This control is achieved by carrying 2 air streams at 2 different temperature levels, one in a *cold* and one in a *warm* duct, to the proximity of mixing units. Each mixing unit proportions air from the 2 ducts in response to the thermostatic control located in its conditioned room and discharges mixed air into the conditioned space at a temperature level as determined by a particular instantaneous internal load. When designed properly, any desired temperature level within the limits of the capacity of the equipment can be maintained in any zone regardless of the load level or load variation in an adjacent zone.

GENERAL DESCRIPTION OF SYSTEMS

On Figs. 1, 2 and 3 are shown the most common arrangements of elementary dual-duct cycles for multi-zone systems. Other arrangements are possible, but thermodynamically they would represent only combinations and variations of these basic cycles.

With the arrangement of apparatus shown in Fig. 1 the resulting average room relative humidity is always somewhat higher than that achieved by operation of the cycles shown on Figs. 2 and 3. Care should be exercised in applying this system to installations requiring a large amount of outside air or where the dew point of the outside air is relatively high.

In the dual-duct system of Fig. 1 the cold air dampers in zone mixing units are functionally equivalent to the face damper of the single-duct system; warm air dampers in the zone mixing units perform the same function as the bypass damper in the single-duct system. Thermodynamically, the 2 systems of Fig. 1 are exactly the same, providing average resulting conditions of the dual-duct are compared to those of the single-duct system. In both cases the average conditions are represented by the state of air in the return air duct. The additional effect of operation of the dual-duct system on the performance of individual zones will be discussed in detail under Performance of Dual-Duct Cycle of Fig. 1.

With the cycle of Fig. 2 it is possible to reduce the moisture content of the out-

* Engineer, Buensod-Stacey, Inc.

Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

side air to any desired level before introducing it into the central system. In this manner variation in average room relative humidity can be considerably reduced, especially under partial load operation. With this cycle, the dew-point of the outside air has no direct bearing on the resulting conditions inside the conditioned spaces.

The analogy of this cycle with a single-duct system as shown in Fig. 2 applies also only when average conditions of the systems are compared. The effect on operation of individual zones will be considered under Performance of Dual-Duct Cycle of Fig. 2.

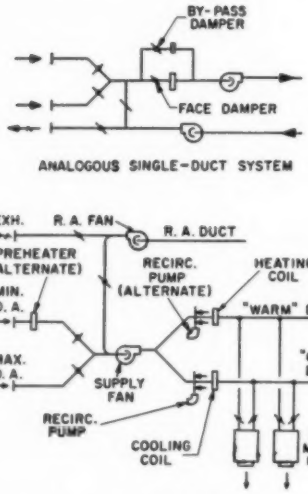

FIG. 1—DUAL-DUCT SYSTEM ANALOGOUS TO SINGLE-DUCT SYSTEM WITH FACE AND BYPASS DAMPERS AND WITH MIXTURE OF OUTSIDE AND RETURN AIR BYPASSED AROUND COOLING COIL

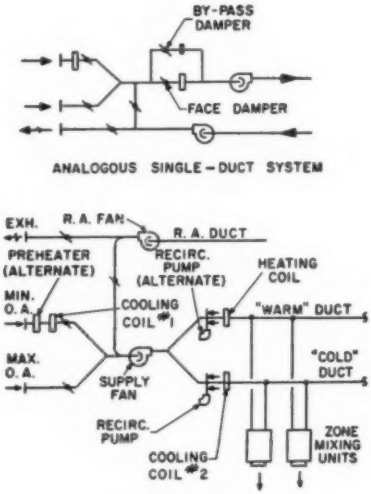

FIG. 2—DUAL-DUCT SYSTEM ANALOGOUS TO SINGLE-DUCT SYSTEM OF FIG. 1 BUT WITH PRECOOLING COIL IN THE MINIMUM OUTSIDE AIR DUCT

The dual-duct cycle of Fig. 3 can bring about minimum variation of relative humidities in conditioned spaces. On summer partial loads when some of the outside air starts to flow into the warm duct the operation of this cycle approaches that of Fig. 1. The disadvantage of this cycle is that it might cause reduction in direct ventilation in zones which require a large amount of warm air unless the warm air temperature is raised as the zone load decreases.

Common to all the cycles shown is the provision for taking in 100 percent outside air when outside conditions are favorable. It should be noticed also that entire conditioning of the air is accomplished in centrally located apparatus, thus relieving the conditioned spaces of any mechanical equipment requiring servicing or cleaning.

PERFORMANCE OF DUAL-DUCT CYCLE OF FIG. 1

Control of the dry-bulb temperature in individual zones is accomplished by room thermostats located in the conditioned spaces. Adequate control of all spaces can be obtained if the temperatures in cold and warm duct are maintained at the proper level and if the total plant capacity is balanced against the imposed loads.

On comfort multi-zone installations, close control of relative humidities is not essential and is never attempted. The principal concern is to avoid excessive humidities by restricting humidity variations to a narrow range resulting in ap-

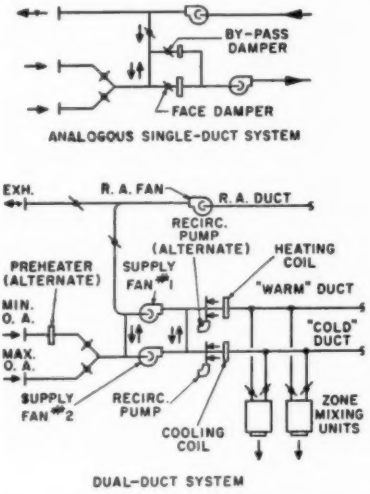

FIG. 3—DUAL-DUCT SYSTEM ANALOGOUS TO SINGLE-DUCT SYSTEM WITH FACE AND BYPASS DAMPERS WHERE ONLY RETURN AIR IS BYPASSED AROUND THE AIR-CONDITIONING EQUIPMENT; TWO SUPPLY FANS

proximately a 1 to 1½ deg variation in effective temperature. The range of 40 to 50 percent in relative humidity has been found adequate for summer operation.

To determine zone humidity level it is necessary always to determine, first, the humidity level in the average zone (represented by the state of the return air) and then, second, the state of the air in individual zones can be calculated. The *no-load* zone indicates maximum variation in relative humidities, and should be used as a criterion to determine the applicability of a particular dual-duct cycle for a given set of conditions.

In Fig. 4 for a given sensible-to-total-heat ratio the average room conditions could be represented by point 1. Point 2 represents the *no-load* zone conditions,

while average conditions shown by point 1 exist in the return duct. Point 2 is at the intersection of line 8-6 and the room dry-bulb temperature. On high pressure systems the heat of compression added to the air by the supply fan is represented by the distance 5-6.

In Fig. 5 the *no-load* zone is again determined by the intersection of line 6-8 and the dry-bulb temperature. Average room condition 3 is obtained by heating the theoretical average mixture of cold and warm air along a new, higher, sensible-to-total-heat-ratio line.

From Figs. 4 and 5 it is apparent that when this system is operating under partial load conditions, the moisture content of the average room air and the *no-load* zone air is higher than under 100 percent load operation. If examination of

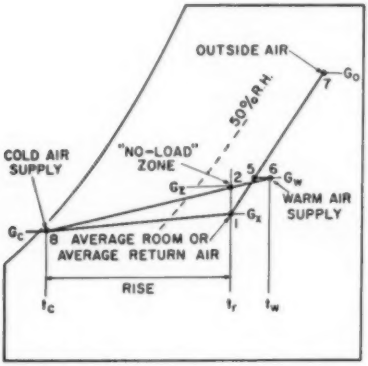

FIG. 4—PSYCHROMETRIC CHART REPRESENTATION OF THE DUAL-DUCT SYSTEM OF FIG. 1

Operating on summer peak load (100 percent internal sensible and latent heat loads) 25 percent outside air.

the cycle on the psychrometric chart were continued by changing conditions one by one and noting the effect produced, it would be seen that room conditions are affected by many other factors in addition to the internal sensible heat illustrated on Figs. 4 and 5.

The factors affecting both the average room conditions and conditions in the *no-load* zone are:

(*a*) Specific humidity of outside air (G_o);
(*b*) Percentage of outside air drawn into the system (P_o);
(*c*) Specific humidity of cold supply air (G_c);
(*d*) Internal latent heat gains (G_R);
(*e*) Percentage of the warm air (P_w);
(*f*) Dry bulb temperature of the warm air (t_w).

To show actual performance of this dual-duct cycle it is necessary to find a general equation to show the combined effect of all the pertinent factors. Con-

sidering only a few of the factors involved could greatly distort the result and not show the true behavior of the cycle. The expression showing the interdependence of all factors affecting the state of the average room air is shown by Equation 1*:

$$G_x = [G_c (1 - P_w) + G_o (P_w P_o) + G_R]/[1 - [P_w (1 - P_o)]] \quad . \quad . \quad (1)$$

A like expression for the state of the *no-load* zone is given by Equation 2:

$$G_z = G_c + [G_x - G_c + (G_o - G_x) P_o] [(t_r - t_o)/(t_w - t_o)] \quad . \quad . \quad . \quad (2)$$

The general algebraic forms of these equations is greatly simplified when used for any particular set of conditions and plotted in convenient graphical form. For

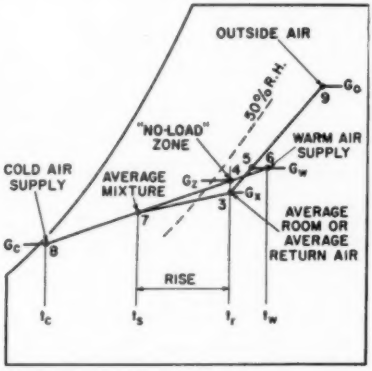

FIG. 5—SAME AS FIG. 4 BUT WITH 50 PERCENT INTERNAL SENSIBLE HEAT LOAD

example, assume it is desired to investigate the behavior of a system with the following set of conditions:

1. Design outside air conditions, 95 F db and 75 F wb, outside air 25 percent of total (G_o = 98 grains, P_o = 0.25);
2. Room 80 F db (t_r = 80 F);
3. Supply air 50 F db and 49.5 F wb (G_c = 51);
4. Initial sensible heat to total heat ratio (SH/TH) = 0.9 (G_R = 5 grains per lb of air handled)

Then, Equation 1 becomes:

$$G_x = 35.3 [(2.11 - P_w)/(1.33 - P_w)] \quad . \quad . \quad . \quad . \quad . \quad . \quad (1a)$$

and Equation 2 reduces to:

$$G_z = 29 + (0.625) G_x \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2a)$$

---

*Derivations of Equations, 1, 2, 1a, 2a, 1b, 2b, 2c, 2d, 3, 4, and 5 are all shown in the Appendix.

Graphical representation of Equation 1*a* is indicated by curves A, B, C, D, and E on Fig. 6, and shows resulting conditions of average room air through the entire range of the cycle, for various arbitrary warm air temperatures. These curves show at a glance the effect of the warm air temperatures on the resulting average room conditions. The effect is relatively small while the system is operating at high loads and is very pronounced when the internal sensible load on a system

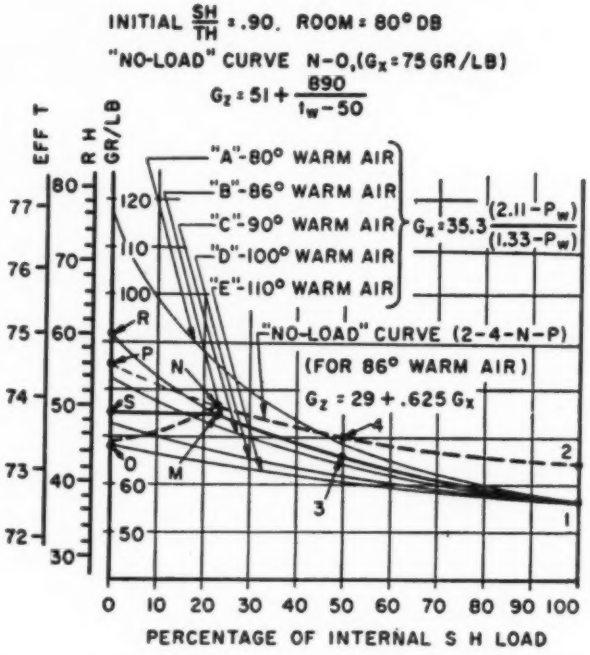

FIG. 6—PERFORMANCE OVER WHOLE RANGE OF INTERNAL SENSIBLE HEAT LOAD OF DUAL-DUCT SYSTEM OF FIG. 1

With 25 percent outside air at 95 F dry-bulb, 75 F wet-bulb, 98 grains per lb; cold air at 50 F dry-bulb, 49.5 F wet-bulb, 51 grains per lb; at various warm air temperatures.

falls to a lower level. This fact immediately gives a clue as to how this system can be controlled to prevent rise of humidity in the conditioned spaces. Various methods of control are possible depending on the medium of heat available and the degree of precision of control of relative humidities desired. As an extreme case, which would result in least variation in the state of the room air, the system could be operated from, say, 100 to 70 percent load at any warm air temperature, say 86 F, allowing a slight rise in room relative humidity, as shown by curve B. At 70 percent load, heat could be applied to the warm air coil raising its temperature to, say, 110 F. This shifts operation from curve B to curve E and keeps varia-

tions of humidities in the room air to a minimum. This method might be feasible and economical when an inexpensive source of reheat is available, such as the hot gas of the refrigerating machine.

On large comfort installations, the following method is suggested. Allow the system to operate without external reheat on a curve such as B up to a point such as M. At that point, apply reheat gradually to the warm air heating coil, which can be accomplished by installing a humidistat in the return duct to control steam flow to the heating coil and, in this manner, limit the rise in average room con-

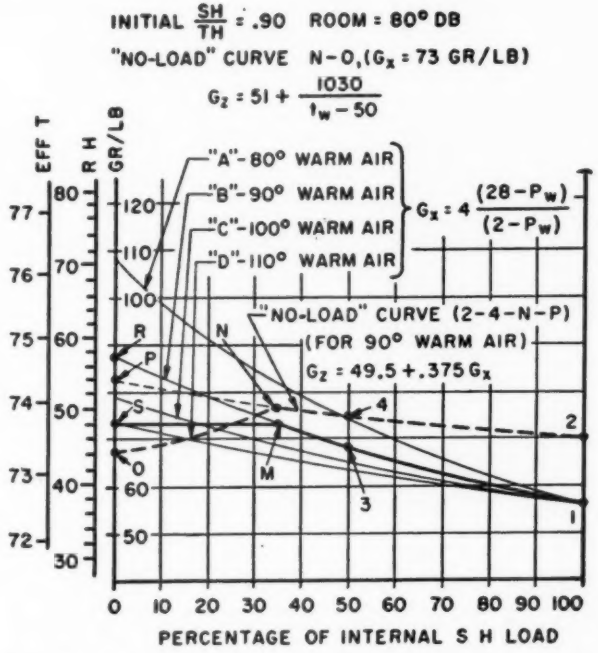

FIG. 7—SAME AS FIG. 6, BUT 50 PERCENT OF OUTSIDE AIR

ditions as indicated by line M-S. While the system as a whole operates on line 1-3-M the conditions in the *no-load* zone could be calculated from Equation 2*a* and shown by curve 2-4-N. From point M down, Equation 2 can be modified as shown in the Appendix and *no-load* zone for constant G_x can be expressed by Equation 2*b*:

$$G_z = 51 + [890/(t_w - 50)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2b)$$

The Equation 2*b* is represented by curve N-O in Fig. 6.

A similar method of representing the performance of a cycle is applied to a system which draws 50 percent outside air, as shown on Fig. 7. It will be noticed

that the system handling 50 percent outside air, in order to prevent rise of relative humidities in the *no-load* zone above 50 percent RH (relative humidity) level, requires a greater amount of reheat than that shown on Fig. 6.

All curves in Figs. 6 and 7 are drawn with the assumption that the outside air dew point temperature remains at its summer design level while the system operates through its entire range. Under these conditions, the curves of Figs. 6 and 7 show maximum possible theoretical variation in the state of room air for the cycle of Fig. 1. In actual operation the lower internal sensible heat conditions coincide with lower outside air dew-point temperatures and the actual operating curves are flatter than those shown in Figs. 6 and 7.

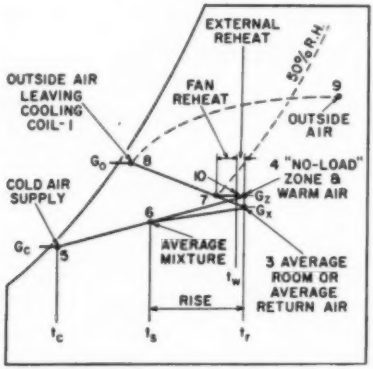

FIG. 8—PSYCHROMETRIC CHART REPRESENTATION OF DUAL-DUCT SYSTEM OF FIG. 2—SUMMER CYCLE—25 PERCENT OUTSIDE AIR—50 PERCENT INTERNAL HEAT LOAD

In order to test the applicability of the cycle of Fig. 1 to a given set of conditions, it is sufficient in most cases to plot only 2 curves, one for average air and one for the *no-load* zone, and then, to determine the point of application of reheat, such as point N on Fig. 6. It is suggested that use of the cycle of Fig. 1 be restricted to conditions when point N falls in the 0 to 40 percent range of internal sensible heat. If a given set of design conditions demanded application of reheat while the system was operating at high load, that would be an indication that the conditions would be better met by a cycle such as shown on Figs. 2 or 3.

PERFORMANCE OF DUAL-DUCT CYCLE OF FIG. 2

The adverse effect on room conditions of the introduction of large quantities of outdoor air at high dew point temperature into air conditioning apparatus can be greatly minimized by the application of the dual-duct cycle of Fig. 2. By installing a cooling coil in the minimum outside air intake the moisture content of the outdoor air can be reduced to any desired level and the rise of humidities in the conditioned

spaces can be restricted on falling loads. However, if the outdoor air is precooled excessively, it may be necessary to apply external reheat to the air in the warm duct even at high loads in order not to overcool lightly loaded zones. Fortunately, on a majority of comfort installations, when the system as a whole, operates at high loads, all zones require some cooling, and partial cooling by *warm* duct is not objectionable.

In Fig. 8 it is assumed that the outdoor air is precooled to a point such as 8. On high pressure systems the mixture of outdoor and return air is reheated by the supply fan to a condition shown by point 10. Additional external reheat as represented by 10-4, is necessary if *no-load* zones are to be satisfied. The average room conditions as shown by point 3 are determined by heating the average mixture of cold and warm air along corresponding average sensible-to-total-heat ratio as

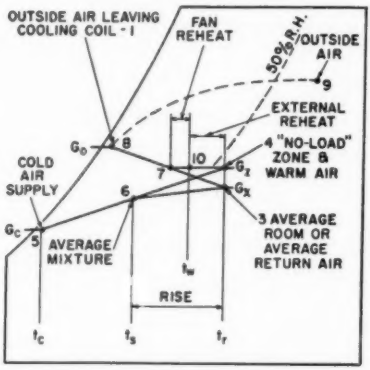

FIG. 9—SAME CYCLE AS IN FIG. 8 BUT WITH 50 PERCENT OUTSIDE AIR

shown by line 6-3. Both *no-load* zone and conditions in the warm duct are represented by point 4.

Performance of the cycle through its entire operating range can be represented again by the general dual-duct Equations 1 and 2 if the meaning of term G_0 is changed to represent the moisture content of air leaving coil No. 1. The operation of such a system through its entire range under a stated set of conditions is shown on Fig. 10. Curves A, B, C and D representing the state of the air in the average room are plotted from:

$$G_x = 42.3\,[(1.76 - P_w)/(1.33 - P_w)]. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1b)$$

The *no-load* zone curve 2-4-N-P for 80 F warm air is shown by:

$$G_z = 19.2 + (0.75)\,G_x \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2c)$$

The *no-load* curve N-O for constant $G_x = 74.5$ grains per lb is represented by:

$$G_z = 51 + [725/(t_w - 50)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2d)$$

Operation of this cycle on comfort installations is similar to that of the cycle of Fig. 1. Assuming that warm air is maintained at 80 F the average room conditions are allowed to rise as shown by curve A up to point M. When the humidity has risen to point M, reheat under control of the humidistat is gradually applied to the warm duct, thus keeping the average room conditions constant (line MS). If the system is controlled in this manner the state of air in *no-load* zones is limited to conditions shown by curves 2-4-N-O.

The conditions obtainable when operating the same system but with 50 percent of outside air are shown on Figs. 9 and 11.

PERFORMANCE OF DUAL-DUCT CYCLE OF FIG. 3

On Fig. 12 Curves A, B, C, D and E show the effect of variation of warm air temperatures on the average room conditions while the system is operating under conditions specified. At high internal loads the warm duct handles only return air and the system is thermodynamically equivalent to a single-duct system with face

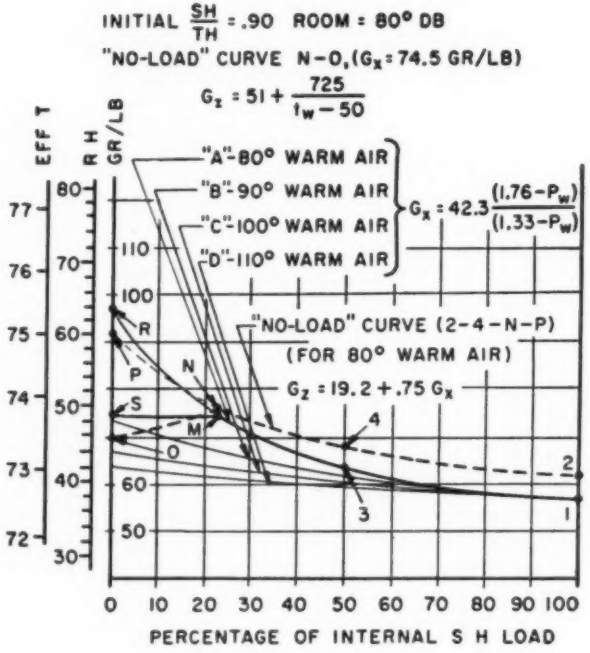

FIG. 10—PERFORMANCE OVER WHOLE RANGE OF INTERNAL SENSIBLE HEAT LOAD OF DUAL-DUCT SYSTEM OF FIG. 2

With 25 percent outside air; air leaving Coil No. 1 at 61 F dry-bulb, 59.5 dew-point, 77 grains per lb; cold air at 50 F dry-bulb, 49.5 F wet-bulb, 51 grains per lb; at various warm air temperatures

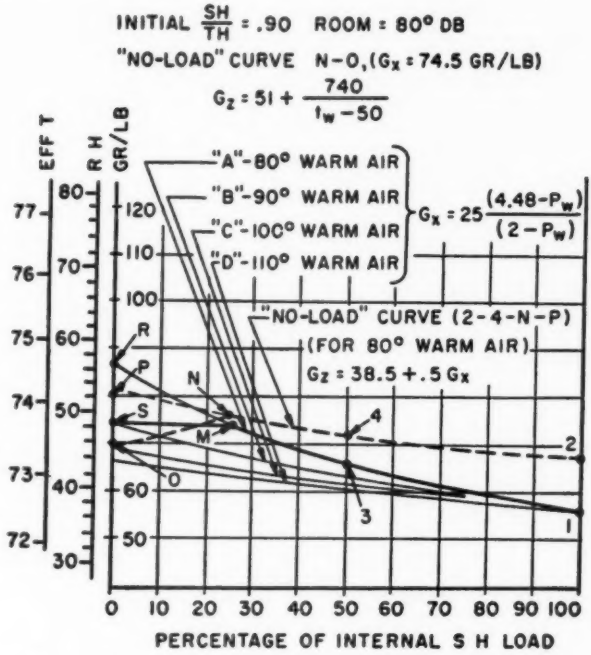

FIG. 11—SAME AS FIG. 10 WITH 50 PERCENT OF OUTSIDE AIR

and bypass dampers where only return air is bypassed around the dehumidifying equipment. At 100 percent load, average room conditions are represented by point 1. All curves from point 1 up to the intersection with line X-X show a gradual rise in humidities and can be expressed by the equation:

$$G_x = 51 + [5/(1 - P_w)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

During this part of the operation the *no-load* zone conditions coincide with average room conditions.

On falling loads when demand for warm air exceeds that of cold air some outdoor air is admitted directly into the warm duct and the performance of this cycle resembles more and more that of the cycle of Fig. 1. The part of performance curves above the line X-X can be represented by equation:

$$G_x = (63 - 4P_w)/(1.5 - P_w) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

It is apparent that, even with this system, reheat must be applied to the warm duct if it is desired to restrict the rise of humidities at low internal loads while high humidity prevails outdoors.

Operating on curve A of Fig. 12 the humidity might be allowed to rise to point M and then reheat would be applied to the warm duct in the manner described for

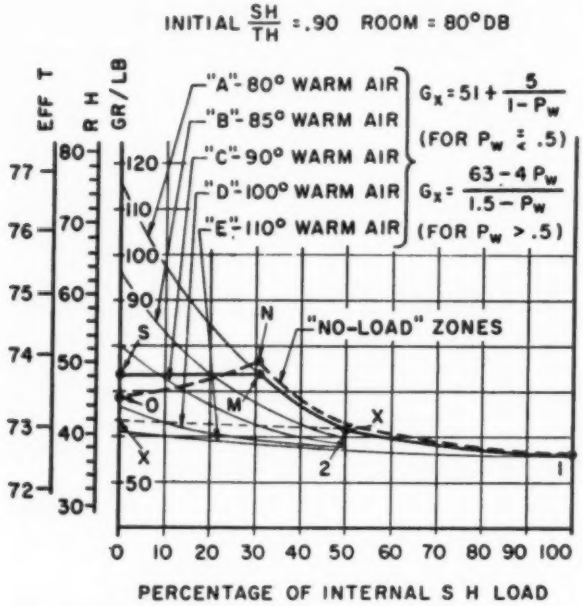

FIG. 12—PERFORMANCE OVER WHOLE RANGE OF INTERNAL SENSIBLE HEAT LOAD OF DUAL-DUCT SYSTEM OF FIG. 3

With 25 percent outside air at 95 F dry-bulb, 75 F wet-bulb, 98 grains per lb; cold air at 50 F dry-bulb, 49.5 F wet-bulb, 51 grains per lb; at various warm air temperatures

Figs. 1 and 2. Gradually increasing the warm air temperature would keep the average room conditions on the humidity level of line MS. For a system operated in this manner, the *no-load* zone conditions are represented by curves 1-2-N-O.

Fig. 13 shows the performance of the same system handling 50 percent outside air. As shown in the Appendix, Equation 3 is applicable for $P_w \leq 0.5$. When $P_w > 0.5$ the temperature curves could be expressed by Equation 5, thus:

$$G_x = 94\,P_w + 14 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

PRESSURE AND AIR VOLUME VARIATIONS

A problem in applying the dual-duct system to large multi-room installations is the operating control of static pressures and volumetric delivery in the distributing system. Due to the nature of the dual-duct system, the diurnal and seasonal variations of internal loads produce constantly changing demands for cold and warm air, thus continuously changing the rate of flow through the dual sides of the distributing system. This change, in turn, causes a wide variance in static pressures in the ducts and upsets the initial volumetric setting of individual outlets serving a

particular area. In all cases the range of pressure and air volume variations reaches a maximum at the end of the run on the last few outlets. Even on smaller systems operating at, say, 2½ in. static pressure on the supply fan, it is possible to have fluctuation of pressures on the last outlet between, say, ½ to 1 in. of water which will cause a change of

$$42 \text{ percent} = \left[\left(\sqrt{\frac{1}{0.5}} - 1\right) \times 100\right]$$

in volumetric delivery to the room being served by the outlet. On larger systems operating, say, at 6-7 in. of static pressure on the supply fan, the static pressure at last outlet might vary between 1 to 4 in. of water changing the rate of air flow into the conditioned space.

$$100 \text{ percent} = \left[\left(\sqrt{\frac{4}{1}} - 1\right) \times 100\right]$$

Variation in air volume of such magnitude may result in a noisy outlet, in faulty air distribution, and in a changed relationship of the controlled room dry-bulb temperature to effective temperature.

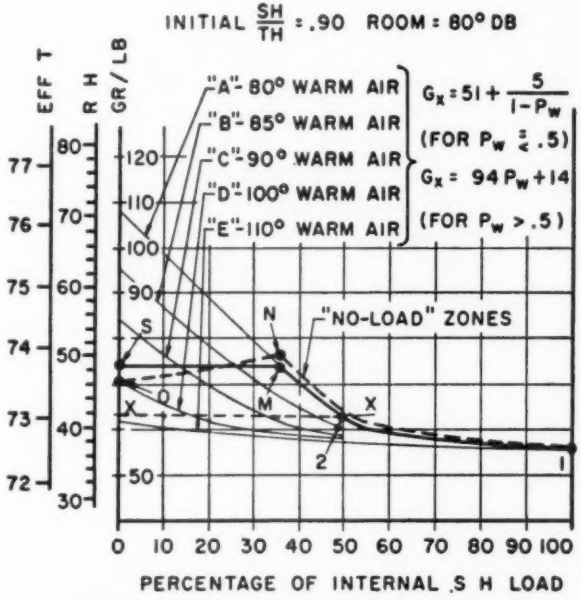

FIG. 13—SAME AS FIG. 12 WITH 50 PERCENT OF OUTSIDE AIR

The basic problem of pressure and volume control can be illustrated by Fig. 14. There the various gradient lines indicate the total and static pressures existing in the elementary system shown at the top. These gradient lines have been plotted for arbitrarily assumed flow in one duct, the total flow remaining constant; namely 100 percent, 50 percent and no flow in either the cold or warm duct.

It is impossible to maintain a constant volume of air flowing through the system with static pressure regulators at the fan or at the entrance to the supply mains. It is assumed, therefore, that the system is equipped with volume regulators placed at each mixing or attenuating unit. This method maintains constant flow to each outlet and consequently constant total fan cfm. Fig. 14 shows the pressure changes occurring throughout the system under normal automatic control and indicates the variations in flow which might occur without effective volume

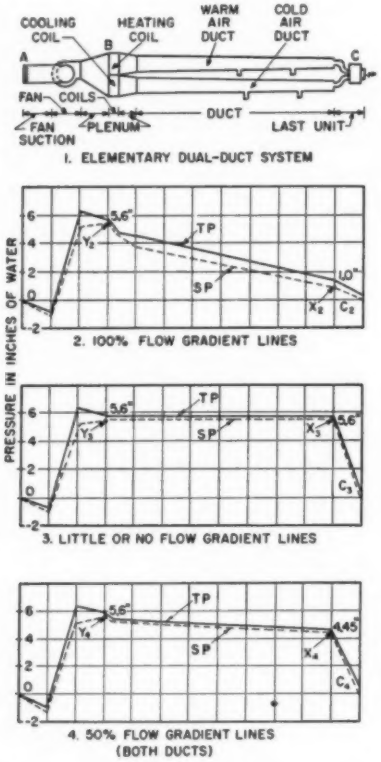

FIG. 14—ENERGY GRADIENTS FOR DUAL-DUCT SYSTEM WITH VARIOUS AIR FLOWS

regulation at the various outlets. Figs. 14-2 and 14-3 represent the pressures in each duct when all air flow is through one duct, a condition approached while the system is operating during summer or winter peak or just after a start-up. These pressures reverse in the 2 ducts when controls shift from cooling to heating or from heating to cooling.

In Fig. 14-2 the last unit is designed for rated cfm at 1 in. static pressure, the duct and coils being designed for 4.6 in. pressure drop (5.6 − 4.6 = 1). If all units, except the last, call for heating and the last unit calls for cooling, the pressure on the last unit will rise to the condition on Fig. 14-3, or 5.6 in. Without effective volume regulation, this last unit would then discharge approximately 236 percent of rated cfm.

If the demand for warm and cold air is equal, the condition of Fig. 14-4 exists. The pressure drop in the duct and coils is now 25 percent = $[(50/100)^2 \times 100]$ and the pressure at the last unit is $[5.6 - (0.25 \times 4.6)] = 4.45$ in. The last unit would then deliver

$$\left(\sqrt{\frac{4.45}{1}} \times 100\right) = 210 \text{ percent}$$

of design cfm.

Without volume control these pressure changes would be different, due to increased flow at the terminal units and to the limits of the fan.

This basic variation in the dual-duct system may be controlled by volume controllers installed as an integral part of each dual-duct mixing unit, which not only keep the air quantity substantially constant in any particular zone, but also maintain a fixed rate of air flow in the entire system, thus preventing fluctuations in power input to the supply fan motor.

ELEMENTS OF DESIGN

The design procedure for a dual-duct system does not differ greatly from the procedure employed in the design of any other multi-room project. The basic principles, procedure and standards are amply presented in ASHAE publications and should be adhered to as closely as possible. In addition, all accepted codes and practices of recognized groups should be incorporated in the design of component parts. However, certain steps in design procedure, as listed here, differ slightly from the established procedure employed in the design of single-duct systems and should be taken into consideration.

1. Since all zones of a dual-duct system handle a substantially constant amount of air at all times, the air requirements for each zone should be checked for demands of summer and winter peaks and for ventilating requirements, and the highest value is used for the zone cfm. Equations 6, 7, and 8 can be used to establish these demands.

$$Q_c^1 = H_s^1/(t_r - t_c)\ 1.08 \quad \ldots\ldots\ldots\ldots \quad (6)$$

$$Q_w^1 = H_s^1/(t_w - t_r)\ 1.08 \quad \ldots\ldots\ldots\ldots \quad (7)$$

$$Q_v^1 = Q_o^1/P_o \quad \ldots\ldots\ldots\ldots \quad (8)$$

2. Total air quantity handled by a supply fan must be equal to the sum of peaks of all zones (as determined by Equations 6, 7 or 8)

$$Q = Q_c^1 + Q_c^2 + \ldots + Q_w^{10} + Q_w^{11} + \ldots + Q_v^{(n-1)} + Q_v^n \quad \ldots \quad (9)^*$$

From Equation 9 it is evident that the total air supply of a large dual-duct system consisting of many zones always exceeds the cold air quantity flowing in the cold duct, even during the summer peak. Similarly during the winter peak the total air quantity is greater than that needed in the warm duct.

3. The quantities of cold and warm air required in a system during summer peak must be determined from Equations 10 and 11.

$$Q = Q_c + Q_w \quad \ldots \quad (10)$$

$$H_s + Q_w\,(t_w - t_r)\,1.08 = Q_c\,(t_r - t_c)\,1.08 \quad \ldots \quad (11)$$

At first glance it might appear that the second term of Equation 11 represents a system penalty on the refrigerating machine. Actually, this is not the case. During summer high load operation no external source of reheat is required in the warm duct and thus the sensible heat being bypassed in the warm duct consists only of the component loads of the system which heat must be removed by the refrigerating system irrespective of the manner in which it is handled. For the cycle of Fig. 1 the second term of Equation 11 consists of the heat of compression of the supply fan as applied to the air flowing in the warm duct plus *outside air reheat* of the portion of outside air being bypassed through the warm duct.

4. The quantities of cold and warm air required in a system during winter peak must be determined from Equations 10 and 12.

$$H_s + Q_c\,(t_r - t_c)\,1.08 = Q_w\,(t_w - t_r)\,1.08 \quad \ldots \quad (12)$$

The second term of Equation 12, again is not a system penalty, provided no refrigeration or evaporative cooling is applied to the air flowing in the cold duct during high load operation. The last term of Equation 12 represents the sensible heat load to be added to the heating coil in the warm duct on peak winter load.

5. The dry-bulb temperatures in cold and warm ducts should be maintained at a level such that at any time it is possible to satisfy the internal sensible loads of the 2 most dissimilar zones with air volumes as fixed by design conditions. Evidently that can be accomplished by an innumerable set of temperature conditions in the cold and warm ducts. In actual design practice the selection of these temperature levels is governed by practical considerations rather than by detailed and theoretical analysis.

During the entire summer cycle the cold air temperature is maintained constant, being selected from the 50 to 55 deg range. For the cycle of Fig. 1, the 50 to 53 deg range is preferable. During winter operation if no deep interior zones were connected to the system, the cold air temperature could be raised by 5 to 15 deg.

The warm air temperature normally is subject to a greater variation than the cold air. In summer during high load operation the warm air temperature as a rule is not controlled and may fluctuate in the 78 to 90 deg range. When the system begins operating at low partial loads, reheat might be added to the warm duct coil raising its temperature to say 95 to 105 deg. For winter operation at peak load the warm air temperature might be selected in the 120 to 140 deg range and, if desired, reduced to a lower level on falling load by an outside air reset thermostat.

APPENDIX

Derivation of Equation 1: Fig. A-1 shows, algebraically, the latent heat conditions existing in a dual-duct apparatus and in the conditioned space.

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* Superscripts in Equation 9 refer to arbitrary zone numbers.

On Fig. A-2 the same conditions are indicated for an analogous single-duct system.

The latent heat flowing into the conditioned space should be equal to that flowing out of the space, or:

$$(1 - P_w)\, G_c + P_w\, [G_o P_o + (1 - P_o)\, G_x] + G_R = G_x P_o + (1 - P_o)\, G_x$$

or rearranging:

$$G_x = \frac{G_c\,(1 - P_w) + G_o P_w P_o + G_R}{1 - [P_w\,(1 - P_o)]}$$

which is Equation 1.

Derivation of Equation 2: Referring to Fig. 5, from the geometry of the cycle it can be written:

$$G_w = G_x + (G_o - G_x)\, P_o$$

$$G_x = G_c + [(G_w - G_c)\,(t_r - t_c)]/(t_w - t_c)$$

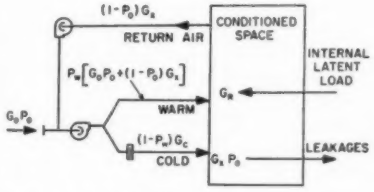


FIG. A-1—LATENT HEAT BALANCE, DUAL-DUCT SYSTEM OF FIG. 1

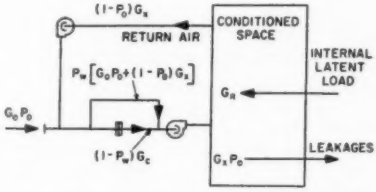


FIG. A-2—LATENT HEAT BALANCE, SINGLE-DUCT SYSTEM OF FIG. 1

combining and rearranging:

$G_s = G_c + [G_x - G_c + P_o\,(G_o - G_x)]\,[(t_r - t_c)/(t_w - t_c)]$, which is Equation 2.

Derivation of Equation 1a: For: $P_o = 0.25$; $G_c = 51$ grains per lb; $G_R = 5$ grains per lb; and $G_o = 98$ grains per lb; Equation 1 can be written:

$$G_x = [51\,(1 - P_w) + 98\,(0.25\,P_w) + 5]/[1 - P_w\,(1 - 0.25)]$$

solving and rearranging:

$G_x = 35.3\,[(2.11 - P_w)/(1.33 - P_w)]$, which is Equation 1a.

Derivation of Equation 2a: For: $P_o = 0.25$; $G_c = 51$ grains per lb; $G_o = 98$ grains per lb; $t_w = 86$ F; $t_r = 80$ F; and $t_c = 50$ F; Equation 2 can be written:

$$G_s = 51 + [G_x - 51 + 0.25\,(98 - G_x)]\,[(80 - 50)/(86 - 50)]$$

solving and rearranging:

$G_s = 29 + 0.625\,G_x$, which is Equation 2a.

Derivation of Equation 2b: For: $G_x = 75$ grains per lb; $P_o = 0.25$; $G_c = 51$ grains per lb; $G_o = 98$ grains per lb; $t_r = 80$ F; $t_c = 50$ F; Equation 2 can be written:

$$G_s = 51 + [75 - 51 + 0.25\,(98 - 75)]\,[(80 - 50)/(t_w - 50)]$$

solving and rearranging:

$G_s = 51 + [890/(t_w - 50)]$, which is Equation 2b.

Derivation of Equation 1b: For: $G_o = 77$ grains per lb; $G_c = 51$ grains per lb; $P_o = 0.25$; and $G_R = 5$; Equation 1 can be written:

$$G_x = [51\ (1 - P_w) + 77\ (0.25)\ P_w + 5]/[1 - [P_w\ (1 - 0.25)]]$$

solving and rearranging:

$$G_x = 42.3\ [(1.76 - P_w)/(1.33 - P_w)], \text{ which is Equation 1b.}$$

Derivation of Equation 2c: For: $G_o = 77$ grains per lb; $G_c = 51$ grains per lb; $P_o = 0.25$; $G_R = 5$; $t_r = 80$ F; $t_o = 50$ F; $t_w = 80$ F; Equation 2 can be written:

$$G_s = 51 + [G_x - 51 + 0.25\ (77 - G_x)]\ [(80 - 50)/(80 - 50)]$$

solving and rearranging:

$$G_s = 19.2 + 0.75\ G_x, \text{ which is Equation 2c.}$$

Derivation of Equation 2d: For: $G_o = 77$ grains per lb; $G_c = 51$ grains per lb; $P_o = 0.25$; $G_R = 5$; $t_r = 80$ F; $t_o = 50$ F; and $G_x = 74.5$ grains per lb; Equation 2 can be written:

$$G_s = 51 + [74.5 - 51 + 0.25\ (77 - 74.5)]\ [(80 - 50)/(t_w - 50)]$$

solving and rearranging:

$$G_s = 51 + [725/(t_w - 50)], \text{ which is Equation 2d.}$$

Derivation of Special Equations for Cycle of Fig. 3, Case I: 25 percent outside air, no direct flow of outside air into warm duct (when $P_w \leqq 0.5$)
Latent heat flowing into conditioned space equals latent heat flowing out of conditioned space

$$P_w G_x + (1 - P_w)\ 51 + 5 = 0.25\ G_x + 0.75\ G_x$$

and rearranging

$$G_x = 51 + [5/(1 - P_w)], \text{ which is Equation 3.}$$

Case II: 25 percent outside air, direct flow of outside air into warm duct (when $P_w > 05$)

$$51\ (1 - P_w) + (P_w - 0.5)\ [(G_x + 98)/2] + 0.5\ G_x + 5 = 0.25\ G_x + 0.75\ G_x$$

and rearranging

$$G_x = [(63 - 4P_w)/(1.5 - P_w)], \text{ which is Equation 4.}$$

Case III: 50 percent outside air, no direct flow of outside air into warm duct (when $P_w \leqq 0.5$)

$$P_w G_x + (1 - P_w)51 + 5 = 0.5\ G_x + 0.5\ G_x$$

and rearranging

$$G_x = 51 + [5/(1 - P_w)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

(same as for 25 percent outside air system).

Case IV: 50 percent outside air, direct flow of outside air into warm duct (when $P_w > 0.5$)

$$0.5\ G_x + (P_w - 0.5)\ 98 + (1 - P_w)\ 51 + 5 = 0.5\ G_x + 0.5\ G_x$$

and rearranging

$$G_x = 94\ P_w + 14, \text{ which is Equation 5.}$$

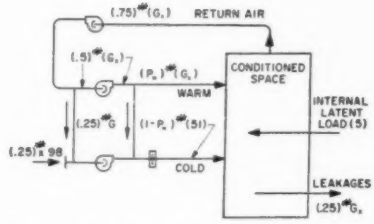


FIG. A-3—LATENT HEAT BALANCE, DUAL-DUCT SYSTEM OF FIG. 3, CASE I

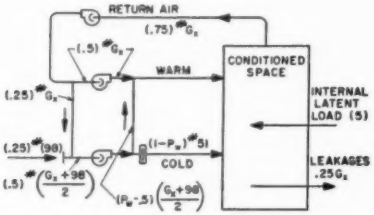


FIG. A-4—LATENT HEAT BALANCE, DUAL-DUCT SYSTEM OF FIG. 3, CASE II

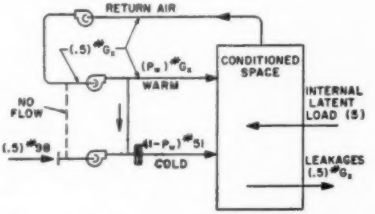


FIG. A-5—LATENT HEAT BALANCE, DUAL-DUCT SYSTEM OF FIG. 3, CASE III

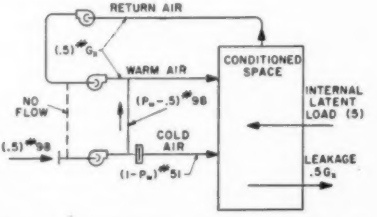


FIG. A-6—LATENT HEAT BALANCE, DUAL-DUCT SYSTEM OF FIG. 3, CASE IV

NOMENCLATURE

G_x = moisture content of average room air, grains per pound.

G_z = moisture content of *no-load* zone air, grains per pound.

G_c = moisture content of cold supply air, grains per pound.

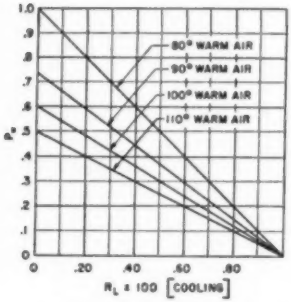
G_o = moisture content of outside air, grains per pound.

G_R = room latent load, grains per pound of air circulated.

G_w = moisture content of warm air, grains per pound.

P_o = ratio of outside to total air.

P_w = ratio of warm to total air $= (1 - R_L)\ [(t_r - t_c)/(t_w - t_c)]$, for $t_2 = 80$ F and $t_c = 50$ F; this equation is represented as follows for various warm air temperatures.



R_L = percentage of total internal sensible heat load divided by 100.

t_r = dry-bulb temperature of room air.

t_c = dry-bulb temperature of cold air.

t_w = dry-bulb temperature of warm air.

H_s = total internal sensible heat load on a system during summer or winter peak, Btu per hour.

H_s^1 = internal sensible heat load of a particular zone during summer or winter peak, Btu per hour.

Q = total air handled by the supply fan, cubic feet per minute.

Q_c = total cold air required by the system during summer or winter peak, cubic feet per minute.

Q_w = total warm air required by the system during summer or winter peak, cubic feet per minute.

Q_c^1, Q_c^2, etc. = cold air required in a particular zone for summer peak, assuming that the zone receives cold air only, cubic feet per minute.

Q_w^1, Q_w^2, etc. = warm air required in a particular zone for winter peak, assuming that the zone receives warm air only, cubic feet per minute.

Q_o^1, Q_o^2, etc. = outdoor ventilating air required in a particular zone, cubic feet per minute.

Q_v^1, Q_v^2, etc. = total air required by a particular zone to satisfy ventilating conditions, cubic feet per minute.

DISCUSSION

R. P. COOK, Rochester, N. Y. (WRITTEN): The author should be complimented for his excellent treatment of a subject which is all too frequently handled in a very loose manner by many design engineers. The paper demonstrates that the analysis and choice of a dual-duct system need not be a difficult and time consuming matter. I know of several instances where either lack or inability of careful calculation have produced results which were not satisfactory and almost impossible of correction.

The installed cost of a dual-duct system is usually greater than that of a single-duct system. Engineers frequently hesitate to employ the dual-duct design, even where it is certain that improved conditions will result, because of the added cost factor. The

designer should be very cautious in his approach at this point and should be completely convinced that the sacrifice does not out-weigh the expense.

A condensation of the information in this paper or at least a reference to the paper could well be incorporated into THE GUIDE.

A. A. GIANNINI, New York, N. Y. (WRITTEN): The author is to be congratulated because this paper is a real contribution to the art of air conditioning system design.

A clear picture is presented of the 2 major design problems of dual-duct systems; namely, wide variations of room relative humidities, and air flow instability. Solutions to both of these problems are presented.

The discussion of pressure and air volume variations is extremely useful because many designers do not seem to understand this problem. The uselessness of static pressure regulation at the fan or at the entrance to the supply mains is adequately demonstrated. In fact such regulation probably exaggerates the instability.

The author shows how humidity variations can be kept within reasonable bounds by 2 different arrangements of system components and he also shows how relative humidity can be kept below a predetermined level by the use of reheat in the warm air duct.

It appears from the various figures that the author uses a room condition of 80 DB and 38 percent RH for design at maximum load. This provokes a number of questions.

(1) Does not this low room humidity add to the refrigeration capacity required at full load?

(2) If a higher room RH is used at full load, would not the need for reheat be increased at partial loads?

(3) Since the use of 80 F room dry-bulb as a design point is rapidly becoming obsolete, wouldn't the paper be more useful to future designers if 75 F or 76 F room temperature had been used in the analysis?

W. A. GRANT, Syracuse, N. Y.: It has been suggested that maybe the midpoint of the system would be a good place to regulate the static pressure. Why is not the duct just ahead of the last unit on the line a better place for the sending tube of the static pressure regulator?

I. A. NAMAN, Houston, Texas: I would like to ask the authors a few questions. One has to do with humidity control where the point was made that in order to properly control the humidity at times of light load reheat would have to be used. I understood this to mean by reheating the cold deck. Wouldn't it be possible to use the heat in the hot deck in order to accomplish the same thing without the additional coils?

The second point has to do with this matter of static pressure control. I believe you will find that there is no place where you could put the static pressure sensing element and have it function properly. It won't work at the end because if you control the end pressure, then the pressure back up the line will rise at low flow. It won't work in the middle and it won't work anywhere else, really. It is only a help. It will reduce the problem, but it won't solve it.

Further along that line, it would seem to me that a proper design and arrangement of the duct system along with a careful analysis of the pressures that will be had, the proper choice of duct sizes, will minimize the static pressure problems and then the further addition of a control might help. I wonder if that has been tried by the authors.

H. E. ZIEL, Detroit, Mich.: Following this paper it was just a little confusing as to why it was necessary to put emphasis on humidity control. It would seem to me that in the cooling cycle the air is drawn down and humidified sufficiently to carry the proper humidity level, especially when it is backheated. I believe that the normal procedure would be to add some heat from the hot duct to raise the dry bulb temperature to the required level after it has been dehumidified in the cold duct at the proper level.

There is one point in the discussion of where to regulate the static pressure in the main ducts. The author did not comment as to the variations that would take place in any air mixing unit with variable static pressures at the inlet. The curve showed that the static pressure at the mixing boxes could vary from 2 to 4 in.

I would like to know what the increase or decrease in the capacity of the air mixing box would be when affected by a change in static pressure at the inlet of the box.

AUTHOR'S CLOSURE, (R. W. Waterfill)*: Any high-velocity system probably costs a little more than a low-pressure auditorium system. However, in multi-room buildings, the central auditorium system is not applicable to exterior zones.

The humidity variations of the dual-duct system compare favorably with any other system of comfort air conditioning. Actually, the humidity automatically remains in the comfort range better with the properly designed dual-duct system than with the central station auditorium system, which is often used for interior spaces of multi-story buildings. Reheat requirements during the cooling cycle are small, equivalent to only a small portion of the internal sensible heat load and exist only when the internal sensible heat load is small. The cost is small compared to the overall operating cost of any system. The reheat may easily be applied to the warm duct and is available from the high side of the refrigerating system.

The low room humidity does require a slightly increased refrigeration plant capacity, an increase similar to increasing the ventilation load from say 25 per cent to 26 per cent of the total air. It also decreases the reheat time for staying within the comfort range. Current trends to greatly increased ventilation and lower room temperatures really do increase plant capacity appreciably. It seems that further design analyses of the dual-duct system, based on lower room temperatures, are in order. Any system of air conditioning which maintains lower room temperatures is going to require a larger plant, both refrigeration and air circulation. In view of the short duration of the maximum outdoor design conditions, however, the value of lower internal conditions should be carefully weighed against increased plant costs.

The problem of air flow instability is one of the major problems, especially in systems where duct runs are long, but I think this problem will be solved. Three methods of flow control have been suggested: (1) temperature manipulation in the supply ducts; (2) static pressure control in the duct; and (3) volume control at each mixing unit. The temperature method might be feasible if the air handling system is oversized and zone requirements are fairly uniform. The static pressure method is a very poor compromise and the only controlled point is at the static tip location, which should be somewhere along the section of main to be controlled. For a main of uniform take-offs, the damper could be about one-third of the controlled length from the beginning and the static tip another one-third farther on. Volume control at each mixing unit is most satisfactory with a reliable device, which can also serve to balance the system.

If the static pressure in a fixed outlet increases from 2 inches to 4 in., the flow would increase 40 percent.

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* Vice President, Buensod-Stacey, Inc., New York.

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No. 1571

BRANCH FITTING PERFORMANCE AT HIGH VELOCITY

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THE PRESENT trend of using higher velocities in air duct systems emphasizes the need for more accurate design data encompassing a greater range of velocities. Over the past few years, considerable data have been collected from laboratory tests of ducts and fittings with velocities up to 4500 fpm. This paper presents an analysis, test results and conclusions based upon laboratory tests of a fitting type of great practical importance—the divided-flow fitting typified by the diagram in Fig. 1. Pressure losses both in the straight-through section of the constant area main and in the branch takeoff are considered.

Analysis of Straight-Through Section

Currently available data for the straight-through section were compared in a recent paper[1] and the need for further research indicated. The lack of agreement between results of different investigators is clearly demonstrated in Fig 1. Since the size and shape of the main is constant, the flow-rate ratio Q_D/Q_U can also be interpreted as the velocity ratio V_D/V_U. The ordinate is the dimensionless *regain coefficient*, which is the ratio of the static pressure rise to the decrease in velocity pressure based upon the mean velocity of flow. In equation form,

$$R = \Delta(SP)/\Delta(VP) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In units of feet of fluid flowing, the loss of total head in the main duct is therefore

$$H = (1 - R)[(V_U^2 - V_D^2)/2g] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Curve A, Fig. 1, is the basis of the present static regain chart[2]. Curve B was obtained from the Carnot-Borda equation for abrupt expansion; namely,

$$H_e = (V_U - V_D)^2/2g \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

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[1] Exponent numerals refer to References.
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Equating 2 and 3,

$$R = 1 - (V_U - V_D)/(V_U + V_D) \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

which can be written

$$R = (2Q_D/Q_U)/(1 + Q_D/Q_U) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

This is the function shown as Curve B in Fig 1. Curve C has been taken from the publication of Konzo[3] and has its origin in a thesis[4]. Curve D was suggested in a recent ASHAE paper[1]. The great diversity in results of different investigators is clearly evident. It is significant that at the higher flow rate ratios, Curves A and C predict much poorer performance than the Carnot-Borda Equation.

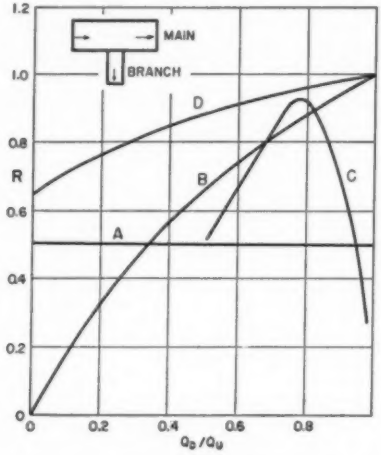

FIG. 1—REGAIN COEFFICIENT DATA FROM FOUR SOURCES

Although Curve C looks radically different from the others, the decreasing values of R at flow-rate ratios above 0.8 have a possible explanation. As developed in the Appendix, the loss can be expressed in terms of upstream velocity heads. Results are shown in Fig. 2. For Q_D/Q_U values above 0.8 it is seen that the Curve C regain coefficients are equivalent to upstream velocity head losses (values of λ_U) of the order of 0.05 or less. Since these tests were conducted with low velocities representative of residential practice, the extremely small pressures involved are very difficult to measure. Hence, a slight discrepancy in this range will have considerable influence on the value of R. However, the net pressure loss is so small that considerable deviation between curves at the upper end are not of serious consequence from a practical standpoint.

As a general consideration of the Fig. 1 curves, from fluid dynamics principles the Carnot-Borda loss given by Curve B should represent the *lower* limit of R. Consequently, laboratory results should ordinarily be expected to fall above this curve. Determination of the *upper* limiting curve of R values requires detailed analysis.

For a one-dimensional frictionless flow, the Bernoulli equation applies and the upper limiting curve is given by $R = 1.0$. For a real fluid, the total head varies from one stream filament to the next because of the distribution of velocity across the section. For the actual case, therefore, it is necessary to determine the upper limiting curve by analyzing a representative flow pattern. Such a pattern is given by the Prandtl Seventh-root Law of velocity distribution[5].

$$v = v_{max}(y/r)^{1/7} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

This relationship is shown in Fig. 3 with the mean velocity V as reference. The distribution of kinetic energy per unit mass across the section is also shown. Near

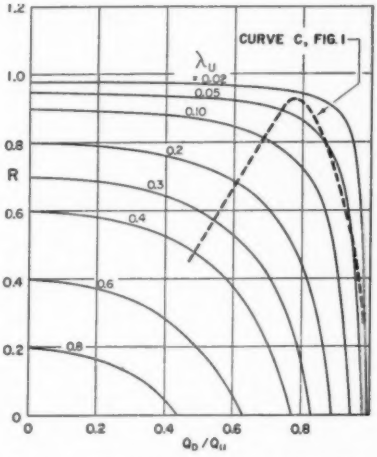

FIG. 2—RELATIONSHIP BETWEEN REGAIN COEFFICIENT, R, AND LOSS IN UPSTREAM VELOCITY HEADS, λ_U

the pipe axis the mean kinetic energy is considerably greater than the kinetic energy of the mean flow (value of 1.0); whereas, near the wall the converse is true. Also, as developed in the Appendix, the total kinetic energy for this velocity pattern is 6 percent greater than that corresponding to the mean velocity of flow. If a type of branch takeoff that removes the flow nearest the pipe wall is visualized, then the flow having the highest kinetic energy continues down the main. As a consequence, more energy than credited with will be available in the straight-through section for conversion to static pressure.

When laboratory tests with air ducts are conducted, regain coefficients are conventionally obtained by dividing the measured static pressure rise by the decrease in velocity pressure as computed from the *mean velocities of flow*. Thus, for air of standard density and velocity units of fpm, Equation 1 is used in the form

$$R = [(SP)_D - (SP)_U]/[(V_U/4005)^2 - (V_D/4005)^2] \quad . \quad . \quad . \quad . \quad (7)$$

Since the numerator is *measured* and the denominator *computed*, excess energy at Section D appears in the numerator as an additional pressure rise which is not accounted for in the denominator. The regain coefficient can therefore be greater than 1.0; *i.e.*, the regain efficiency can be greater than 100 percent. It should be emphasized that R values exceeding unity do not violate the conservation of energy law. Such values occur only because Equation 7 does not account for the nonuniform distribution of kinetic energy across the pipe section.

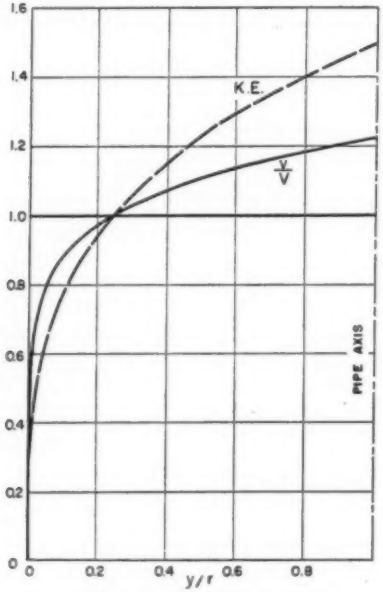

FIG. 3—DISTRIBUTION OF VELOCITY AND KINETIC ENERGY WITH A SEVENTH-ROOT PROFILE

To obtain a curve of maximum attainable R values, the flow configuration of Fig. 4 will be analyzed under somewhat idealized and simplified conditions. Presume the flow into the branch is taken from the periphery of the pipe, as indicated by the flow boundary line, so that the Section 2 flow continues along the main. Also, presume the velocity pressure corresponding to the mean velocity of the Section 2 flow is conserved and therefore converted to static pressure at Section D. Under these conditions, the relationship between regain coefficient and flow-rate ratio can be determined analytically. The development is given in the Appendix. With a seventh-root profile at Section U, the result is

$$R = \{[1/(1 - x_0)^4] - 1\}(Q_D/Q_U)^2/[1 - (Q_D/Q_U)^2] \quad . \quad . \quad . \quad . \quad (8)$$

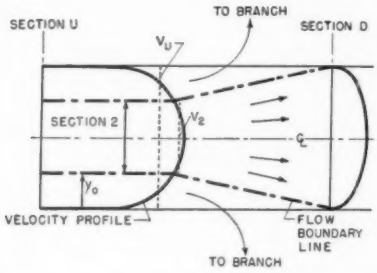

FIG. 4—FLOW CONFIGURATION ANALYZED

in which

$$Q_D/Q_U = 1 + (8/7)\, x_0^{15/7} - (15/7)\, x_0^{8/7} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$0 \leq x_0 \leq 1$$

Curve E of Fig. 5 applies to one-dimensional, frictionless flow and has the constant value unity. Curve F represents Equation 8, and it is to be noted that R is always greater than 1.0. The curve therefore confirms the foregoing reasoning that regain efficiencies above 100 percent are physically possible.

Equation 6 applies for Reynolds numbers up to about 50,000. Values considerably in excess of this are often attained in high velocity duct systems. For the

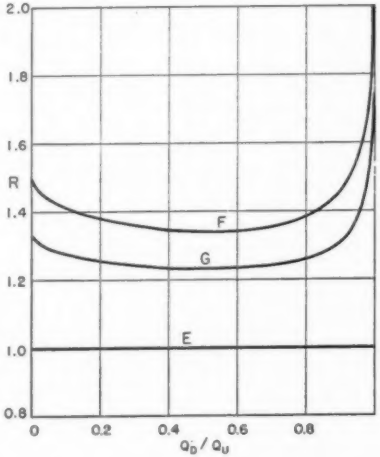

FIG. 5—RESULTS OBTAINED ANALYTICALLY

highest velocities and largest ducts, velocity profiles are better represented by the tenth-root relationship[8].

$$v = v_{max} (y/r)^{1/10} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The resulting function is shown by Curve G of Fig. 5, and its equation is given in the Appendix. Although the shape of G is similar to F, the R values at corresponding flow-rate ratios are less. This would be anticipated because a tenth-root profile is flatter than a seventh-root profile and, as a consequence, the kinetic energy is more uniformly distributed over the pipe cross section. Note that if the exponent on y/r in Equation 10 is decreased to zero, the one-dimensional case (Curve E) is obtained.

The preceding analysis and curves show that, when computing test results by Equation 7, static pressure regain efficiencies above 100 percent are physically possible without violating the conservation of energy.

COMPARISON WITH TEST RESULTS

Results of laboratory tests with a 3-in. diameter branch duct connected to an 8-in. diameter main are shown by the circles and Curve H in Fig. 6. Following conventional laboratory practice[1], the regain coefficient was computed from Equation 7 with the measured pressures corrected for friction losses between the junction and the upstream and downstream measuring stations.

Curve G is included for comparison and it is notable that the curve representing experimental data follows the same trend, and also that values of R above 1.0 are obtained from test data. The squares are results from tests with other divided-flow fittings of various area ratios and designs. Since each design was studied over a narrow range of flow-rate ratios, no attempt has been made to fit curves to the results. Their primary purpose is to show that R values above 1.0 were also obtained with other fitting designs. Curve B represents the Carnot-Borda expansion loss and has been transposed from Fig. 1; this curve should represent the lower limit of data collected.

Study of Equation 7 will show that as $Q_D/Q_U = 1.0$ is closely approached the value of R will be sensitive to very slight inaccuracies in laboratory measurements. Therefore, the last plotted point on Curve H is not considered firmly established, and the curve has been drawn as a broken line. No great difficulty is introduced, however, because values of Q_D/Q_U above 0.95 rarely occur in the design of air duct systems.

There are 2 principal reasons why Curve H is below Curve G. The former includes energy dissipation, whereas the latter is based on essentially no loss of mechanical energy. Secondly, the actual branch duct, being simply butted to the side of the main, does not draw its fluid from the entire low energy periphery of the main flow. Instead, it takes its flow from the region around the branch opening so that only a portion of the flow of lowest kinetic energy is drawn into the branch, the remainder being taken from the high energy flow in the central region of the pipe.

From the analysis and test results, it is concluded that Curve H of Fig. 6 is rational and is representative of performance attainable with conventional divided-flow fittings when utilized for medium and high velocity applications.

APPLICATION CONSIDERATIONS

Main ducts of high velocity systems in commercial buildings frequently have several branch ducts connected at various intervals. Also, the branch ducts are frequently of the same size and resistance and have the same design flow rate. Any unbalance in the system is primarily the result of the friction pressure loss in the main duct between takeoffs. It is, therefore, very desirable that the flow hav-

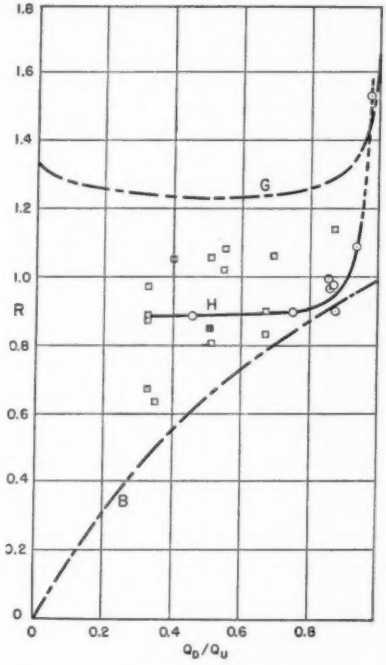

FIG. 6—COMPARISON OF ANALYTICAL AND EXPERIMENTAL RESULTS

ing the highest energy be available in the straight-through section for overcoming this friction loss. The characteristic indicated by Curve H is therefore advantageous from the design standpoint. Moreover, a fitting design which closely approached the Curve G performance would tend to minimize system unbalance in most applications.

Since mean velocities and static pressure are conventionally used as a basis for system design, Curve H has been used to formulate the static regain chart of Fig 7. With design velocities V_U and V_D, the chart yields values of the increase in static pressure between Sections U and D. Comparison of values with those in the litera-

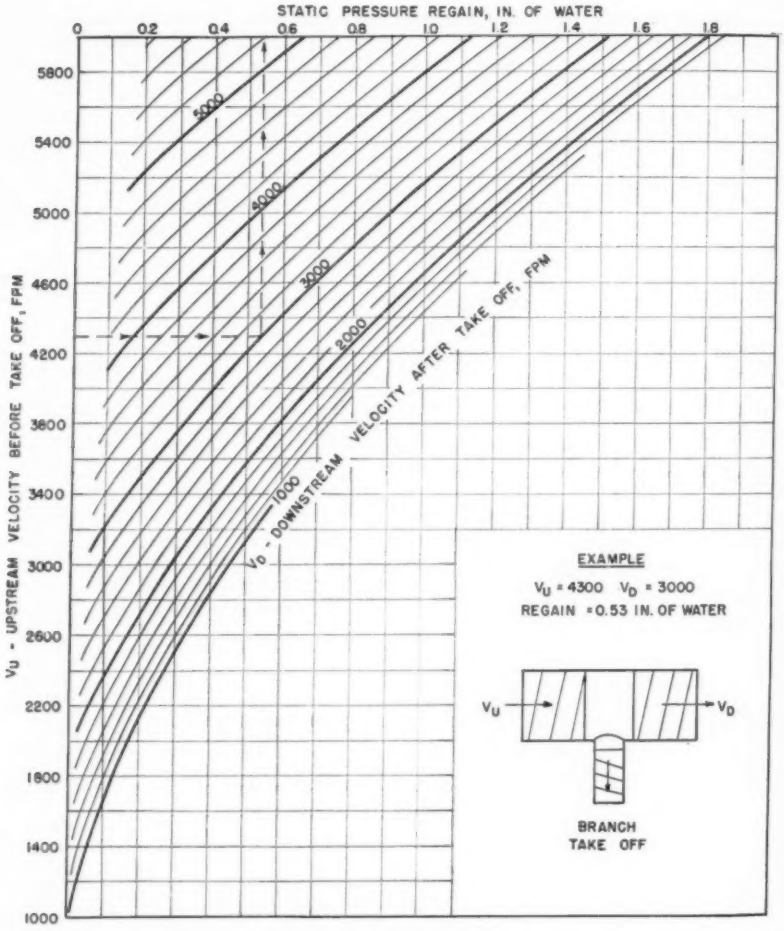

FIG. 7—STATIC REGAIN CHART

ture[2] show that much more static pressure is available downstream of the takeoff than heretofore presumed. The marked increase in effectiveness is seen by comparing Curve H with the present regain basis of $R = 0.5$ in Curve A. The resultant increase in available pressure is also in agreement with field tests in which static pressures toward the ends of main ducts have been found to be higher than anticipated. This new chart is considered to contain much more realistic and authoritative design information. It is hoped the chart will contribute to the improvement of air duct design methods, particularly for high velocity systems which are becoming increasingly numerous.

The size of the main duct is frequently reduced after a branch takeoff, and it is desirable to know the static pressure in the smaller duct. The reducing fitting connecting the ducts usually has an included angle of 45 to 60 deg, and the pressure loss is very small. A representative value is 5 percent of the velocity head in the smaller duct[6], which is approximately 3 equivalent diameters.

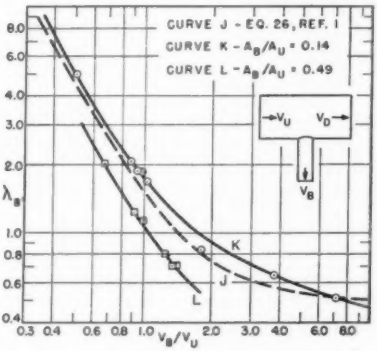

FIG. 8—EXPERIMENTAL AND PREDICTED RESULTS FOR CYLINDRICAL BRANCH TAKEOFFS

The principal factor is the conversion of static pressure to velocity pressure as the flow accelerates. The static pressure will decrease by the difference in velocity pressures after and before the transition. A chart is available for determining velocity pressures from known values of mean velocity[2]. The static pressure at the beginning of the smaller duct can be found by *adding* the amount of regain from Fig. 7 to the static pressure at Section U and *deducting* the change due to energy conversion.

DIVERTED-FLOW SECTION

As pointed out recently[1], the pressure loss in the branch takeoff must be considered separately from that in the straight-through section. Concerning the flow diverted into the branch, test results are ordinarily expressed in terms of a dimensionless loss coefficient λ_B. This coefficient is the loss in total pressure divided by the velocity pressure corresponding to the mean velocity of the branch flow[1]. Test results for 2 takeoffs of the conventional type are presented in Fig. 8

and compared with Curve J, a predicted relationship for small branch-to-main area ratios[1]. Curves J and K are in good agreement. Curve L falling below Curve J represents the effect of area ratio, and its relative position is in agreement with results in the paper by Korst[7]. It should be remarked that the Curve L area ratio is much larger than ordinarily used in practice.

From an application standpoint, static pressure is frequently used as the design basis. Therefore, these same results are presented in Fig. 9 with the ordinate λ_{SP} denoting the loss in static pressure in terms of branch velocity heads. Corresponding Curves are K-N, J-M and L-P. Note that for velocity ratios less than 1.0, the curves based on static pressure are very sensitive to small differences be-

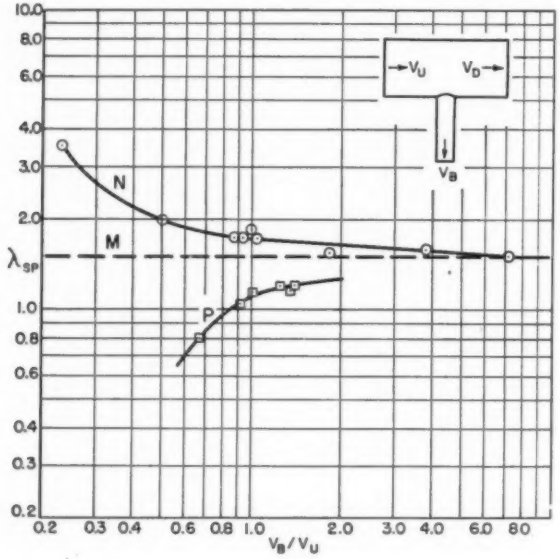

FIG. 9—RESULTS FOR CYLINDRICAL TAKEOFFS WHEN EXPRESSED ON STATIC PRESSURE BASIS

tween the λ_B curves of Fig. 8. Also, Curve M predicts the loss in static pressure is independent of velocity ratio; this was not pointed out in the source[1] and may have important theoretical or practical significance.

A conical takeoff has been shown to be effective in reducing the takeoff loss[1]. This is confirmed in Fig. 10 by the test results shown by Curve R. Comparison with Curve M, which applies to the type diagrammed in Fig. 9, shows the pressure loss is reduced to less than one-third at low velocity ratios. Curve S was obtained by converting available data on conical takeoffs to the static pressure basis. The lower loss for Curve S as compared with Curve R is a result of the larger ratio of inlet to branch duct areas.

The conical takeoff is very effective because it provides a nozzle type of entrance to the branch, whereas the cylindrical takeoff provides an orifice type of entrance. Separation of the flow from duct surfaces and resultant stream contraction are thereby reduced, and the pressure loss during re-expansion considerably lessened.

From the analysis and test results for the diverted-flow it is concluded that the experimental data are in substantial agreement with works of others, and that the use of a conical takeoff will reduce takeoff losses considerably. For takeoffs of about the same configurations as those illustrated in Fig. 10, Curves R and S can be used directly for design data.

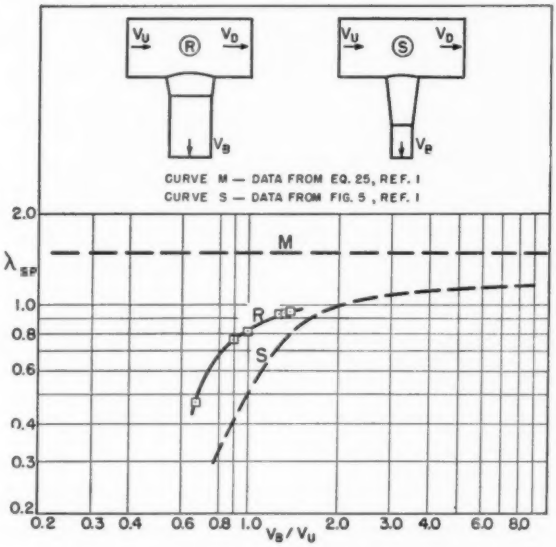

FIG. 10—RESULTS FOR CONICAL TAKEOFFS

CONCLUSIONS

1. Analysis of the conventional method of correlating static regain test data shows that regain coefficients greater than 1.0 can be obtained.

2. Experimental data on the straight-through section follow the trend of the analytical relationships developed, and are considered representative of performance attainable in the field.

3. The static regain chart presented constitutes an improvement in design information available for medium and high velocity applications.

4. Results of branch takeoff tests correlate well with those of previous investigators.

5. Pressure losses of conical types of branch takeoffs are considerably less than

those of cylindrical types. Information presented can be used directly as design data.

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SYMBOLS

A = area.
d = differential of a function.
H = loss of head, ft of fluid flowing.
R = regain coefficient defined by Equation 7, dimensionless.
(SP) = static pressure, inches of water.
(VP) = velocity pressure, inches of water.
Q = volume rate of flow.
r = radius of pipe.
y = radial distance from pipe wall.
y_0 = radial distance from pipe wall to flow boundary line (see Fig. 4).
x = dimensionless radius, y/r.
x_0 = dimensionless parameter, y_0/r.
v = velocity.
V = mean velocity over a cross section, Q/A.
λ = loss coefficient (pressure loss divided by a reference velocity pressure), dimensionless.
ρ = mass density, slugs per cubic foot.

Subscripts:

U = upstream cross section of main duct.
D = downstream cross section of main duct.
B = branch duct cross section.
e = Carnot-Borda loss.
max = maximum.

APPENDIX

Development of λ_U *Curves in Fig 2:* Equation 2 was divided by the upstream velocity head to obtain

$$H/(V_U^2/2g) = (1 - R)\,(V_U^2 - V_D^2)/V_U^2 \quad . \quad . \quad . \quad . \quad . \quad \text{(A-1)}$$

Designating the left side by λ_U and solving for R, there results

$$R = 1 - \lambda_U/[1 - (V_D/V_U)^2] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-2)}$$

Since the area of the main duct is constant, it can be written as

$$R = 1 - \lambda_U/[1 - (Q_D/Q_U)^2]. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-3)}$$

Selected values were assigned to λ_U in Equation A-3 and the curves in Fig. 2 plotted from the equation.

Total Kinetic Energy of Flow: The increment of cross sectional area is

$$dA = 2\pi (r - y)\, dy \quad \text{(A-4)}$$

and from Equation 6 the velocity is

$$v = v_{max} (y/r)^{1/7} \quad \text{(A-5)}$$

Writing the continuity equation for constant density in the form

$$\pi r^2 V = \int v\, dA. \quad \text{(A-6)}$$

and substituting the first 2 equations yields

$$\pi r^2 V = \int_0^r v_{max} (y/r)^{1/7} 2\pi (r - y)\, dy \quad \text{(A-7)}$$

For convenience, the integral is expressed in dimensionless form by setting

$$y/r = x \qquad 0 \leq x \leq 1$$

so that

$$dy = r dx$$

and Equation A-7 becomes

$$\pi r^2 V = 2\pi v_{max} r^2 \int_0^1 x^{1/7} (1 - x)\, dx \quad \text{(A-8)}$$

Integration yields

$$v_{max} = 1.225\ V \quad \text{(A-9)}$$

Substituting this in Equation A-5, the velocity profile in terms of the mean velocity is

$$v = 1.225\ V (y/r)^{1/7} \quad \text{(A-10)}$$

The total kinetic energy over a cross section is given by

$$E = \rho/2 \int v^3\, dA \quad \text{(A-11)}$$

With Equations A-4 and A-10 this becomes

$$E = \rho/2 \int_0^r (1.225\ V)^3 (y/r)^{3/7} 2\pi (r - y)\, dy \quad \text{(A-12)}$$

Following the procedure above, the dimensionless form is

$$E = \rho/2\ (2\pi r^2)\ (1.225\ V)^3 \int_0^1 x^{3/7} (1 - x)\, dx \quad \text{(A-13)}$$

Integration yields

$$E = 1.06\ [\pi r^2 \rho\ V^3/2] \quad \text{(A-14)}$$

The bracketed term is the kinetic energy based upon the mean velocity of flow, E_m. Hence,

$$E = 1.06\, E_m \quad \text{(A-15)}$$

Thus the actual total kinetic energy of a seventh-root velocity profile is 6 percent greater than that computed from the mean velocity of flow. By a similar analysis with Equation 10, the excess is found to be 3 percent for a tenth-root profile.

Development of Regain Coefficient Curves: For the conditions stated in the text, the pressure rise is

$$(SP)_D - (SP)_U = (V_2/4005)^2 - (V_D/4005)^2 \quad \text{(A-16)}$$

Substituting in Equation 7 and recalling that $A_U = A_D$, there is obtained

$$R = [(V_2/V_U)^2 - (Q_D/Q_U)^2]/1 - (Q_D/Q_U)^2 \quad \text{(A-17)}$$

From Fig. 4, the flow rate at Section 2 is seen to be

$$Q_2 = \pi\,(r - y_0)^2\, V_2 \quad \text{(A-18)}$$

which can also be written

$$Q_2 = \pi r^2\,(1 - x_0)^2\, V_2 \quad \text{(A-19)}$$

For Section U a similar expression is

$$Q_U = \pi r^2\, V_U \quad \text{(A-20)}$$

Dividing Equation A-19 by Equation A-20 and recalling that $Q_2 = Q_D$, there is obtained

$$V_2/V_U = [1/(1 - x_0)^2]\,(Q_D/Q_U) \quad \text{(A-21)}$$

which, when substituted in Equation A-17 yields

$$R = \{[1/(1 - x_0)^4] - 1\}\,(Q_D/Q_U)^2/[1 - (Q_D/Q_U)^2] \quad \text{(A-22)}$$

This is Equation 8 in the text.

Next, the flow rate ratio is related to x_0. Noting Equation A-8, the corresponding form for the Section 2 flow is

$$Q_2 = 2\,\pi\, v_{max}\, r^2 \int_{x_0}^{1} x^{1/7}\,(1 - x)\,dx \quad \text{(A-23)}$$

Substituting Equation A-9 and integrating, the result is

$$Q_2 = \pi r^2\, V_U\,[1 + (8/7)\, x_0^{15/7} - (15/7)\, x_0^{8/7}] \quad \text{(A-24)}$$

Utilizing Equation A-20, and again recalling that $Q_2 = Q_D$, the final form is

$$Q_D/Q_U = 1 + (8/7)\, x_0^{15/7} - (15/7)\, x_0^{8/7} \quad \text{(A-25)}$$

This is Equation 9 in the text.

A generalized form of Equation A-25 can be obtained by expressing the velocity profile as

$$V = v_{max}\,(y/r)^{1/n} \quad \text{(A-26)}$$

and repeating the foregoing development. The result is

$$Q_D/Q_U = 1 + (1 + 1/n)\, x_0^{2+1/n} - (2 + 1/n)\, x_0^{1+1/n} \quad \text{(A-27)}$$

The applicable relationship for the tenth-root profile given by Equation 10 can therefore be written immediately. It is

$$Q_D/Q_U = 1 + 1.1\,x_0^{2.1} - 2.1\,x_0^{1.1} \quad \ldots\ldots\ldots\ldots \quad \text{(A-28)}$$

This and Equation 8 in the text were used to obtain Curve G of Fig. 5.

DISCUSSION

PROFESSOR W. G. COLBORNE, Kingston, Canada: The four curves of Fig. 1 are used to show a general disagreement between different investigators. The 4 curves shown do not strictly apply to the illustrated take-off as drawn on Fig. 1. Curve A for example is the basis of the static regain chart in THE GUIDE and applies mainly to cases where the main is reduced in size following the branch. Curve B on the other hand applies to a straight duct with an abrupt expansion which may or may not be similar to a main and branch set-up. If there is a similarity between the conditions with which the paper deals, and the conditions on which these curves are based, then I feel it should be more fully explained.

Curve C, I assume, is the plot of the performance of one fitting or one group of similar fittings. In Fig. 2 this curve C is superimposed on curves of various values of λ_u. I don't think a given fitting would exhibit constant values of this term λ_u for different values of Q_d/Q_u in fact the paper has later shown that this λ_u could be zero or even negative. I do not therefore see the purpose of Fig. 2 in attempting to justify the drop in curve C.

Although the heading of the paper is for high-velocity work I wonder why the theory or basic thinking presented would not apply to normal velocity work. The root profile theory would be modified for various Reynold's numbers but this would not change the possibility of obtaining values of R equal or greater than one.

I think it would be helpful to include in the paper the actual values of velocities used in the tests.

This excellent paper presents the results of sound fundamental thinking and experimental evidence to back up this thinking. Our hard working committee on air distribution now have the proof that a revision is sorely needed in the static regain section of THE GUIDE. There is much more work of this sort required; however, Dr. Gilman and his associates deserve much credit for this fine paper.

P. A. ARGENTIERI, Danbury, Connecticut: I just wanted to mention how much I appreciate seeing a paper of this type incorporated into the literature because there has certainly been a great need for more accurate data on static regain.

In this regard I would like to ask the author whether or not the use of the conical as against the cylindrical takeoff would further increase the static regain given in the chart, which I assume applies to the standard cylindrical takeoff shown in the illustration. Presumably the conical take-off, which allows the high-energy flow in the main duct to progress without interruption would further increase the static regain.

Another question that I have, is what is the actual geometry of the conical take-off and how critical is this?

W. O. HUEBNER, New York, N. Y.: The authors should be congratulated. This is a fine, useful paper, and I hope that the information which it contains will be incorporated into THE GUIDE at the earliest possible moment.

I would like to ask the authors if they will give us another paper containing information on duct take-offs where the branch forms not a 90 deg but a smaller angle with the main duct.

AUTHORS' CLOSURE (DR. GILMAN): With regard to Professor Colborne's questions, Curve A is the basis of the present static regain chart in THE GUIDE and it *does* apply to

the fitting type illustrated in Fig. 1. The regain charts are based on such a fitting, and the slight pressure loss in the reducing section following it is ignored.

Curve B is simply the representation of the Carrot-Borda equation for abrupt expansion. The development is given in Equations 3, 4 and 5.

Curve C is based on considerable data from University of Illinois research. Many different sizes and shapes of branch takeoffs were studied and reported in References 3 and 4. The curve is a representative average curve for the data.

Incidentally, in Reference 1 it is shown that rectangular and round branch takeoff data correlate well when geometries are not widely dissimilar. Moreover, correlation on a dimensionless basis was attained with results of different investigators which included water as well as air.

Curve D was suggested in Reference 1 as a possible good correlation curve.

The lines of constant λ_u in Fig. 2 were calculated by the procedure given in the first part of the Appendix. By superimposing Curve C from Fig. 1, it is clear that above $Q_D/Q_U = 0.8$ the pressure loss in terms of upstream velocity heads (λ_u) is very slight. The point here is that Curve C is very sensitive to slight discrepancies in instrument readings in this range. As a consequence, the radical falling off of Curve C does not mean that gross errors exist in the Illinois test data.

As to the velocities used, the maximum was 4500 fpm. The lower limit was about 800 fpm. Since variations in flow rate ratio Q_D/Q_U are effected by throttling the flow, the range of velocities and pressures in the system is necessarily wide.

Professor Colborne is correct in stating that the theory itself is applicable to medium as well as high-velocity systems. In these tests, attention was directed to obtaining application data for high-velocity systems. His complimentary remarks on the paper are gratefully acknowledged.

In reply to Mr. Argentieri, the cylindrical type of takeoff undoubtedly disturbs the straight-through flow to some extent. The conical takeoff therefore has the potential of increasing the static regain somewhat. However, in Fig. 6, the upper theoretical limit given by Curve G is fairly closely approached by the test points, leading to the conclusion that the increase in regain would be slight with a conical takeoff. Collection of test data would be desirable.

The conical takeoffs and main ducts shown in Fig. 10 are circular and drawn to scale. The area ratios are therefore the ratios of the squares of the linear dimensions. The included angle for both is approximately 20 deg. The pressure loss curves in Fig. 10 show that Fitting S has the more favorable geometry. At low-velocity ratios, the geometry is seen to have considerable effect on the loss.

In reply to Mr. Huebner, the information on takeoffs at various angles would be of considerable value. We do not have immediate plans for investigating other phases of the problem. Since the variables to be studied would include branch size, velocity ratio and angle of discharge, the subject might well be investigated by the ASHAE Laboratory or a University under the guidance of the TAC on Air Distribution. There is much to be learned concerning the fundamental nature of the flow at such junctions.

1572

No. 1572

SELF-ACTUATED ROOM CONTROL FROM HIGH-SPEED AIR

By EDWIN F. SNYDER, JR.*, MINNEAPOLIS, MINN.

THE COST of individual space control with high velocity air-handling systems is rather high due to the number of units normally involved in this type of system. It is desirable to reduce this cost as much as possible by reducing the amount of electric wiring or pneumatic piping required if this can be done without sacrificing control.

High-velocity air is a source of readily available energy. A static pressure of 1 in. water column (WC) is equivalent to 0.03609 lb per sq in. or 0.23 lb per sq ft. When operating with a 6-in. static pressure there is available a pressure of 31.38 lb per sq ft. Presence of this large amount of energy in the flowing air suggests that it be made the basis of accomplishing this desirable goal of cost reduction.

APPARATUS AND INSTRUMENTATION

The test work involved in the evaluation of this system included 2 general phases. The first covered the performance of the controls and air valve while the second phase was devoted to the evaluation of the noise levels of the system.

The equipment used for the control performance tests consisted of: 2 draft gages (0–10.0 in. water), one draft gage (0–0.50 in. water), one draft gage (0–0.25 in. water), a 3½ in. calibrated orifice, a high pressure blower and duct system, a dial indicator (0.0005 in.), and a pitot-static tube.

The noise evaluating tests made use of the following facilities and equipment: one sound control room, one high pressure blower and duct system, one sound-level meter (Type 1551A), one octave-band analyzer (Type 1550A), a set of calibration equipment, and a Rochelle-salt crystal microphone.

In order to properly evaluate the sound level of the equipment being tested, it was necessary to have a space in which background and extraneous noises were held to a minimum. For this purpose, a sound test room was constructed and all laboratory tests were conducted in this room.

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* Supervisor, Control Application Commercial Division, Minneapolis-Honeywell Regulator Co. Member of ASHAE.

Presented at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

PROCEDURE

The control of a high-pressure air-handling system must consider 3 basic factors, *viz*: (a) air temperature, (b) air velocity; and (c) noise.

A fourth factor might be considered, namely static pressure, but since this is so directly related to air velocity, it is not included in the list.

All high-velocity air-handling systems can be divided into 2 basic types; namely the single-duct and the dual-duct. In the single-duct system, the amount of heating or cooling accomplished in the controlled space can be varied by varying the temperature of a fixed volume of air or by varying the volume of air held at a fixed temperature or by a combination of these 2 methods. The dual-duct system normally utilizes one duct for carrying chilled air and the other duct for carrying heated air. The 2 ducts terminate in a mixing box where the air is combined in the proper proportions to meet the requirements of the confined space.

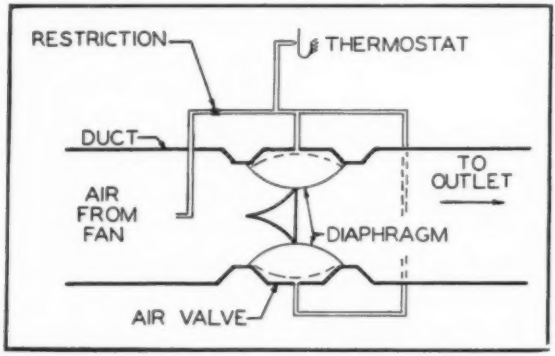

FIG. 1—SCHEMATIC OF VALVE TYPE RESULTING FROM PRELIMINARY DESIGN WORK

Since the dual-duct system appears to be the more complex from a control point of view, this is the system that was used as the design target. The considerations outlined in the introduction to this paper and an analysis of the characteristics of the dual-duct high-pressure system indicated that a self-actuated control system would be practical.

Preliminary design work provided a type of air valve such as that shown in Fig. 1. The material for the flexible diaphragm was a critical item since it must be highly flexible under all operating conditions of temperature, must have a high fatigue level, must resist abrasion and must not deteriorate over a long period of time. Such a material was found and operating models were built on the order of the unit shown in Fig. 1.

The unit makes up one control system and the operation is as follows: The flexible diaphragms each form one side of an air chamber whose other side is rigid. The chamber is tapped for an air connection which is, in turn, connected through a tee-connection to a bleed-type room thermostat and then terminates in a pressure

pick-up head upstream of the air valve. A restriction is placed in the air line between the pressure pick-up head and the thermostat.

Referring again to Fig. 1, it is seen that the pressure pick-up head is exposed to duct pressures on the upstream side of the air valve. This is the point of highest pressure in the segment of the system shown. There is a pressure drop across the air valve due to the resistance of the valve and the thermostat is mounted in the controlled space where the pressure is atmospheric.

Assuming that the duct shown in Fig. 1 is carrying heated air, then as the space temperature rises, the bi-metal thermostatic element moves against the nozzle and closes it. The air entering the pick-up head under the pressure at that point is transmitted to the air valves. Since the pressure upstream from the valve is higher than at the valve, this causes the flexible diaphragms to move out against

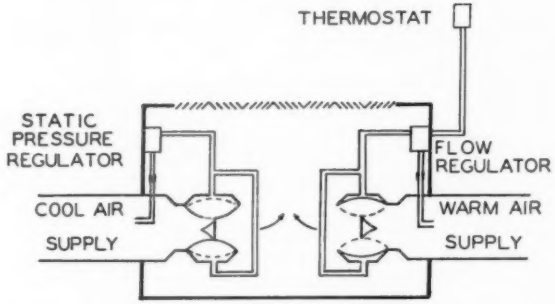

FIG. 2—SCHEMATIC AIR VALVE FOR USE WITH DUAL-DUCT SYSTEM WHEN DUCTS TERMINATE IN A MIXING CHAMBER

the center stops and thus, closes the duct. As the space temperature falls, the bi-metallic sensing element moves away from the thermostat nozzle and air is allowed to bleed out through the nozzle. Since the restriction is between the pick-up head and the thermostat, air can be bled faster through the thermostat nozzle than it can be supplied through the restriction and the flexible diaphragm collapses. If the nozzle is wide open, the diaphragm assumes the approximate position shown by the dotted line at the air valve. If the nozzle is not opened wide, the diaphragm assumes an intermediate position as determined by the pressure balance, and proportioning action results.

Fig. 2 shows schematically a dual-duct unit where the ducts terminate in a mixing chamber. In this case, one duct carries chilled air and one heated air. Investigation shows that the warm air duct is controlled in the same manner as the system in Fig. 1. The only difference lies in the fact that a flow regulator has been added. The flow regulator acts very much like a relay and increases the effective capacity of the system thus reducing the system response time.

It will be noted that there is no inter-connection of controls between the warm air and cold air ducts. Installations of this dual-duct type normally require a constant volume of air delivered to the space and that pre-supposes a constant static pressure within the mixing chamber. Thus, if there is an increased call

for heat so that the warm air duct opens there must be a corresponding drop in cold air supplied to the mixing chamber or the discharge volume cannot be held constant. The system shown indicates a static pressure regulator controlling the air valve on the cold air duct. As the static pressure within the mixing chamber varies due to a repositioning of the warm air valve by the thermostat, the static pressure regulator varies the flow through the cold air valve so that the static pressure within the mixing chamber remains relatively constant.

Fig. 3 shows the air delivery from a dual-duct system and mixing chamber where the control system was calibrated for a delivery of 325 cfm from the mixing cham-

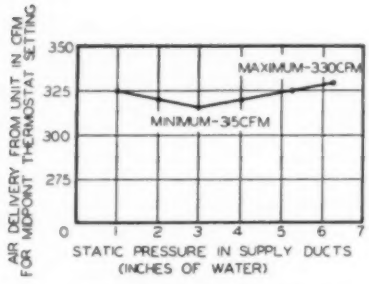

FIG. 3—VARIATION OF AIR DELIVERED FROM DUAL-DUCT MIXING CHAMBER WITH SUPPLY DUCT PRESSURE CHANGES (TEST RESULTS WITH CONTROL CALIBRATED FOR 325 CFM DELIVERY)

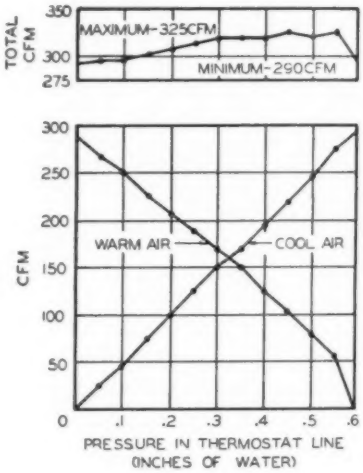

FIG. 4 (*Right*)—TEST RESULTS LIKE FIG. 3 BUT WITH CONTROL CALIBRATED FOR 290 CFM DELIVERY

ber. It will be noted that the delivery from the mixing chamber varied from 315 cfm to 330 cfm as the static pressure in the supply ducts varied from 1 in. water to 6.25 in. water.

Fig. 4 shows the results of another test run on the same system but with the control system calibrated for a 290 cfm delivery from the mixing chamber. The curve at the top of the figure shows the total air delivery as the demand goes from all heating to all cooling.

The curves on the bottom of Fig. 4 show the deliveries from the cold air and hot air ducts as the demand changes from 100 percent heating to 100 percent cooling. It will be noted that, due to the relay action of the flow regulator, the pressures in the thermostat line are considerably less than in the supply ducts.

Fig. 5 shows a means by which both air ducts with their separate valves can be combined in a single Y-type unit. The figure shows an air flow silencer or sound attenuator added to the outlet of the Y-type unit. A unit of this general type is suitable for use with ceiling type outlets where other mixing chambers are not necessarily available.

The Y-type unit was installed on an actual job in the field. Performance test results are shown in Fig. 6.

THE NOISE PROBLEM AND THE SOUND EVALUATION METHODS

An extremely critical factor in any high pressure air handling system of this sort is the noise level of the system. Since controls of the type discussed here are an integral part of the complete system, it is essential that their noise level be such

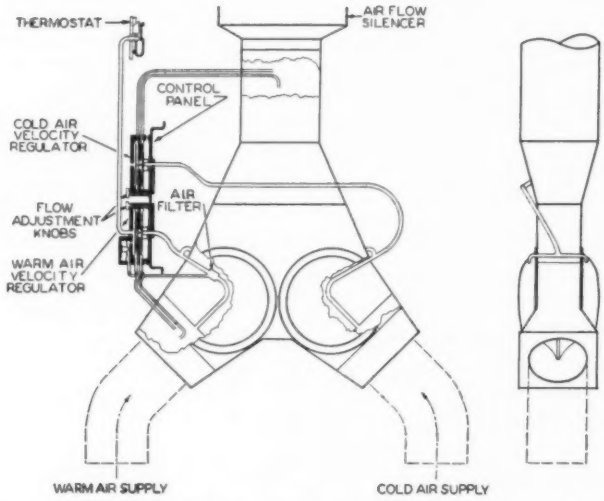

FIG. 5—SCHEMATIC SHOWING A MEANS OF USING SEPARATE AIR VALVES AT "Y"-CONNECTION FOR COMBINING DUAL-DUCT AIR STREAMS

that it will not increase the overall effect to too great an amount. Space permits only a few brief remarks concerning noise and its measurement.

Noise, briefly, is an undesirable sound. This sound was measured by a sound level meter which measures sounds of essentially 3 different types: *i.e.*, it has 2 filter weighted networks (A and B scale) to approximate human ear response and it has a flat response (C scale) which is also used to measure actual sound pressure levels. Proper application of the instrument permits measurement of sound pressure level which can be used to determine the actual acoustic properties of a sound source.

Frequencies of the sound components were studied with the assistance of the octave band analyzer—an instrument which allows the insertion of various filter networks into the measuring system so as to cause the exclusion of all but one particular band of frequencies. The scale of frequencies runs in 8 octaves from 37.5 to 9600 cycles per second (cps). Although the scale on the analyzer and the scale

often discussed in theory is 2–75 cps in the first band and 4800–10 Kc in the last band, the actual response of the instrument and a more exact consideration of the term octave band gives the range of frequencies stated.

In analyzing a noise source, the particular frequency (in this case a band of frequencies one octave wide) causing the noise as well as its sound pressure level in relation to the other frequencies becomes very important. This fact is due mainly to the frequency response characteristics of the human ear. Not only do different frequencies affect a given ear in a particular manner, but people also differ in response in connection with various noises under various conditions. This is, of

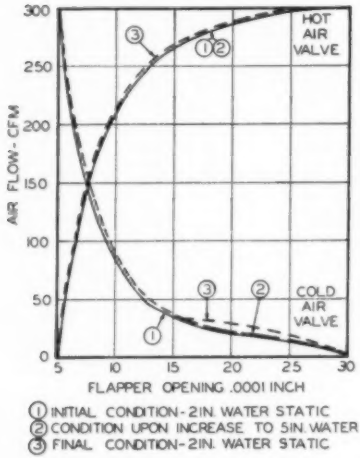

FIG. 6—FIELD TEST RESULTS WITH VALVES INSTALLED BY METHOD SKETCHED IN FIG. 5

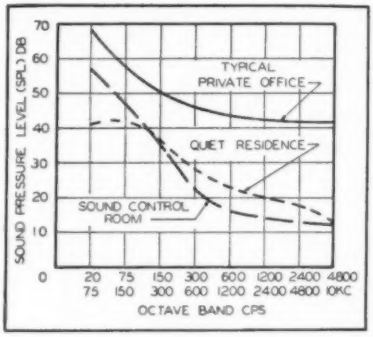

FIG. 7—SOUND PRESSURE LEVEL AT CENTER OF SOUND CONTROL ROOM AND SOUND LEVELS FOUND IN A PRIVATE OFFICE AND A QUIET RESIDENCE

course, the complicating factor with which any noise evaluation technique must be concerned.

Although several methods of noise evaluation have been developed, the method used in the sound test laboratory is that proposed by Armour Research Foundation. This method begins, as in all methods, with the sound pressure level in each octave band. The band pressure level is then plotted on a chart developed especially for the purpose of evaluating noise characteristics. The plotted points are then connected by a smooth curve for convenience and the loudness of each band is read on the sone scale appearing on the chart. Addition of the band loudness through the 8 octaves gives the total loudness of the device under test.

In the case of a grille or opening discharging air, it can be said that the sound source is directional and that the sound being radiated represents a random frequency spectrum. However, the source can be treated somewhat more simply by neglecting, for the moment, its directional characteristics. Further, since in this

case the unit was to be measured in an enclosure (the sound control room) certain conditions had to be established. The essential consideration is that the microphone should not be placed in the area of a standing wave where the walls of the enclosure are causing a considerable increase in sound pressure due mainly to reverberation in the room. The problem can be simplified further by establishing a test code which provides microphone positions based on comparative results rather than absolute acoustic properties. Ambient sound characteristics had to be established and this was done by using current methods and with due regard to recommended corrections such as those of the *American Standards Association* (ASA) which require that there be at least 2 db difference between the ambient level and the level observed with the test device in operation. Table 1 represents corrections to be applied when the difference is less than 10 db.

TABLE 1—CORRECTION FOR AMBIENT SOUND LEVEL

Difference in db between ambient sound level and sound level with device under test operating (total sound level)	2	3	4	6	8	10
Correction to be applied to total sound level, decibel	−4.3	−3.0	−2.2	−1.3	−0.8	−0.4

The sound measurement test room used is essentially an enclosure capable of providing the required ambient sound level and having the necessary accessories to deliver a known quantity of air to the device under test. Construction is designed to attenuate the sound level of the surrounding area in general, and to reduce as much as possible the structural vibrations present. The inner room which is the essential element of the whole system, consists of a room floating on isolation pads with walls, floor and ceiling of masonry material. All entries for air control and electricity are of a flexible nature to increase the isolation characteristics. The inner surfaces of the control room are lined with a highly absorbent material 3 in. thick. The air supply into the control room is metered through either of the 2 supply ducts, according to *ASME* flow measurement procedures, with a 5-in. sharp-edged orifice in an 8-in. duct. The ambient characteristics of the control room are as follows: The sound level ranges from 25 to 30 db measured according to the method previously described; the sound pressure level at the speech interference level is 12 to 17 db; and the loudness level is approximately 1.5 sones over the full frequency spectrum. The range of sound level or sound pressure level is due mainly to a low frequency rumble present in the structural system and will be high or low depending on what particular equipment is operating in the basement of the test laboratory. These ambient characteristics permit noise measurements to be made on a majority of the devices which would be normally considered as noise generators. Fig. 7 indicates the relative sound pressure level of various specified locations, one of which is the approximate geometric center of the sound control room.

NOISE EVALUATION OF "Y"-TYPE DUCT CONTROL UNIT

Now then, to proceed with the noise evaluation of the "Y"-type duct. The first phase of the investigation was to determine the loudest condition of air delivery; *i.e.* (*a*) the loudness, in sones, at 1 in. static and 300 cfm total flow and (*b*) the loudness at 6 in. static and the same total flow. In order to provide data which

might later be used on a comparative basis, it was decided that the test code of the *Industrial Unit Heater Association* be followed with regard to microphone location. The unit under test was installed in the center of the sound control room with a horizontal, upward, discharge 3 ft from the floor. This location allowed measurements to be made in the near field of the source. Fig. 8 represents an evaluation of the data wherein the total loudness for condition (*a*) is found to be 12.3 sones

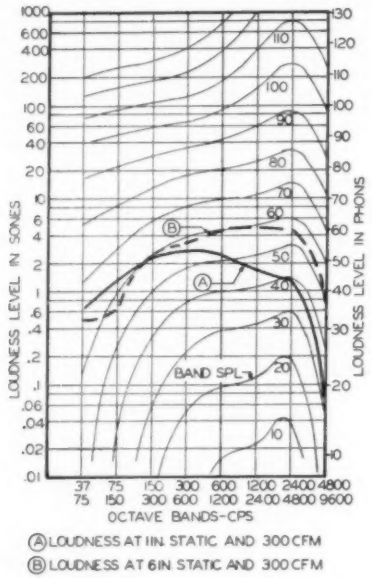

FIG. 8—LOUDNESS EVALUATION USING "Y"-TYPE MIXING UNIT

and the loudness for condition (*b*) is found to be 21.5 sones. It is well to mention here that during the testing it was found that the total flow from the device which is stated to be 300 cfm might possibly be in error due to ductwork leaks between the flow meters and the "Y"-type unit. However, this defect in test procedure will have little effect on a relative basis.

The second phase was to determine which frequency band made the greatest contribution to the overall loudness of the device. Also, since this particular octave band varies with static pressure, to compare the results of condition (*a*) and (*b*). Condition (*a*) has its greatest portion of loudness in the 300 to 600 cps band and condition (*b*) was loudest in the 1200 to 2400 cps band. This information is found on Fig. 8.

Sound tests were conducted on the same "Y"-type unit with a sound muffler or attenuator attached.

The "Y"-type valve together with the muffler was installed in the sound control room at an angle of about 45 deg to the floor. The location and angle were chosen so as to provide the maximum amount of distance between the discharge of the muffler and the test enclosure. A search of the sound pressure level in a plane through the nozzle or discharge opening of the muffler at a 1-ft radius revealed that the spread of pressure was less than 2 db.

The mike was placed at the position of greatest sound pressure level found in the discharge plane. A similar search in the plane through the opening of the "Y"-type valve at a distance of 1-ft from the edge of the opening revealed that no position was greater than other positions by more than 4 db. The mike was also placed in the position of greatest sound pressure level when measuring the sound being produced by the "Y"-type valve. Investigation revealed that these positions were not located in a standing wave and they were approximately into the diffuse field.

TABLE 2—SUMMARY OF OBSERVED DATA SHOWING ACTUAL BAND LEVEL ATTENUATION OBTAINED BY ADDITION OF THE MUFFLER

BAND, CPS	SPL WITHOUT MUFFLER, DB	SPL WITH MUFFLER, DB	ATTENUATION, DB	
			DESIRED	ACTUAL
75–150	73	60	7	13
150–300	76	57	10	19
300–600	73	45	22	28
600–1200	67	34	30	33
1200–2400	65	34	34	31
2400–4800	58	28	34	30
4800–9600	51	16	32	35
Noise Rating	44.5 sone	5.8 sone		

Table 2 is a summary of the observed data and also lists the actual band level attenuation obtained by the addition of the muffler.

Reference to Table 2 will reveal the 37.5–75 cps band has not been reported. This is due to the fact that there is not enough difference between the sound pressure level air delivery and the ambient sound pressure level. This elimination of the low band pressure level affects the noise rating of the device. However, the magnitude of this effect can be neglected in this case since the low band pressure level without the muffler contributed approximately 1 sone to the noise rating while the contribution with the muffler was probably less than 1 sone. It is well to mention that the ambient level was measured with no air being delivered to the device under test. Comparing the noise input to the muffler with the noise output it is found that the muffler reduced the noise by 87 percent.

An actual installation of the "Y"-type units and the individual units, such as shown in Fig. 2 was made in an office and both air delivery and noise performance checks were made. The results of the flow tests were very nearly the same as those obtained in the laboratory tests. The noise tests gave an equally satisfactory result.

ACKNOWLEDGMENT

The author would like to acknowledge with sincere thanks the contribution made to the sections dealing with noise by S. O. Gullicksen of the Minneapolis-Honeywell Regulator Co.

CONCLUSIONS

The performance tests conducted in the laboratory and reported herewith demonstrate that it is possible to design an air mixing unit to be used on high velocity (high pressure) air distribution systems that will:

1. Deliver a reasonably constant total air flow while the duct pressures vary between 1 in. water and 6.25 in. water.
2. Vary the mixing ratio of hot and cold air supply through a range of 0 percent to 100 percent respectively.
3. Use the available air duct pressure to actuate the automatic control device for temperature and air flow control.

The laboratory tests also provide information on the noise levels developed and the attenuation effectiveness of the mufflers used with the air mixing mechanism.

DISCUSSION

C. M. ASHLEY, Syracuse, N. Y.: I would like to ask the author one question. Where was the sound level reading taken with respect to the first reading, that is the reading before the attenuator; and then after the attenuator?

What I would like to emphasize is that in evaluating effective performance of equipment, it is very desirable to use sound power level, rather than sound level as such, because then we can interpret the results in terms of the room level, but it would be very helpful if we knew where the sound pressure level was taken; then we could make our own evaluation.

H. H. YOUSOUFIAN, New York, N. Y.: My comments are in reference to the data presented in Fig. 8. In Fig. 8 the author presents the loudness level for the various octave bands for the unit tested and, by way of reference, representative sound pressure levels are also given in Fig. 7 for a private office and a residence. My question is whether the raw test data are presented in Fig. 8, or has it been corrected for the *live room* effects of say a residence or private office so that a direct comparison may be made with the curves given on Fig. 7?

AUTHOR'S CLOSURE: Mr. Ashley asked a very good question with regard to the location of the microphone in the sound tests. The location was just beyond the discharge grille. There were several locations taken. The ones referred to, as I recall were about a foot beyond the center of the discharge grille. We did work with sound pressure levels. I did not take the time to go into that here today, but you are so right. It is necessary to do that for complete comparison and it was done. There is a better discussion in the paper, incidentally.

Mr. Yousoufian, the question between the 2 figures I am sorry that I didn't point that out. The results of the curve shown in Fig. 8 are based upon tests run in the same sound room, not under exactly ideal conditions in that they don't represent exactly experimental data from the actual field installation, but for comparative purposes we thought it would be better at this time.

1573

No. 1573

ENVIRONMENTAL PHYSIOLOGY

By R. W. KEETON*, M.D., CHICAGO, ILL.

IT IS with great humility that I accept the F. Paul Anderson Medal. In so doing, I wish to emphasize the large contributions of my associates in the Departments of Medicine and Mechanical Engineering of the University of Illinois to the formulation and completion of our studies. The award emphasizes the importance which the Society has come to attach to *Environmental Physiology* as a field for investigation, the results of which are of great value to the general public.

F. Paul Anderson was an educator at heart, a faculty early recognized in his career by his appointment to the deanship of the School of Mechanical Engineering of the University of Kentucky, one year after he joined the engineering faculty. Later he became dean of the College of Engineering. His vision that engineering schools should teach fundamentals, combined with his great personal drive in support of his convictions secured recognition for Kentucky's College of Engineering. Thirty years after joining the staff of the University of Kentucky he became the first director of the Research Laboratory of the ASHVE. Under his directorship the experiments in the field of *Environmental Physiology* established the comfort chart and effective temperature. The practical importance of these studies was at once widely recognized. In 1927 Bazett[1] summarized critically the studies on Physiological Responses to Heat. He reproduced the charts showing the effective temperature scale for human subjects fully clothed and stripped to the waist, and indicated that this was another important approach to the measurement of the cooling power of the atmosphere. From the Society's point of view these studies represented a turning point in its research interests. The Society now recognized that in addition to physical problems involving equipment and process air conditioning that they were also confronted with the physiologic problems of comfort air conditioning involving effects of machine-made atmospheric environments on people.

We may now briefly turn to some of the problems in *Environmental Physiology* which have been studied and to others projected.

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[1] Exponent numerals refer to References.

* Professor Medicine Emeritus, University of Illinois, College of Medicine. Member ASHAE.

Presented on the occasion of the fourteenth award of the F. Paul Anderson Medal to Dr. Keeton at the 62nd Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Cincinnati, January 1956.

ENTERING AND LEAVING SHOCK

The distressing sensations produced in individuals entering from the hot humid street a space conditioned for comfort, or leaving the comfortable environment and returning to the hot humid street, have been designated as *entering and leaving shock*. Because these sensations were so distressing laymen and many physicians were convinced that such exposures were dangerous to health. Since 1950 there has been an increasing tendency in designing air-conditioned spaces to abandon the differentials previously proposed for outside and inside air in summer and winter. Our studies at the University of Illinois[2] have shown that these transitions from widely different environments are made with physiological ease and without danger even to individuals with compensated heart disease. For well people there is no need for being concerned about indoor-outdoor temperature differentials either summer or winter. W. M. Vidulich, ASHAE Public Relations Director, has aptly summarized the studies by the caption, *Keeping Cool Won't Harm You.*

EFFECT OF COMFORTABLE ENVIRONMENT ON INDUSTRIAL PRODUCTIVITY

It would seem that a comfortable worker should be more productive than an uncomfortable one. Discomfort introduces certain distracting psychological factors which are in abeyance in comfortable conditions.

There is a growing feeling in industry that, when large groups of workers are observed over long periods of time, their unit production is higher under comfortable than under uncontrolled environmental conditions. This impression is reflected in the striking increase in installations of air conditioning in office buildings. However, it is difficult to secure objective evidence to support the validity of this impression or to clarify the mechanism responsible for the improvement. The study of the effect of the environment on production requires a knowledge of the physiology of muscular activity and especially the mechanism of muscular contraction. Once this is understood, the effects of loads imposed by the atmospheric environments on muscular activity can be measured.

The origin of the energy for muscular contraction, its transfer, and its restoration, had been studied intensively from 1920 to 1930. An agreement on the present concept of the physico-chemical changes involved was summarized by A. V. Hill[3] in 1932. The energy for the contraction of muscle is derived primarily from the dissociation of phosphocreatine (phosphagen). In the presence of oxygen resynthesis of phosphagen occurs readily. Normally a deficit of oxygen rapidly develops in the contracting muscle. The energy for the resynthesis of phosphagen must be obtained from a reaction which can proceed under anaerobic conditions. This is found in the formation of lactic acid from the muscle glycogen. The extent of the oxygen debt is indicated by the quantity of lactic acid present. With these chains of reaction occurring within the oxygen deficient muscle cell the H and OH ions may become unbalanced with an accumulation of lactic and other acids to the point at which both the chemical reactions and the muscle contractions are slowed. The process by which the composition of the fluids within the muscles and the lymph are maintained so that such reactions as previously described can proceed smoothly, is defined as homeostasis. Once homeostasis becomes unbalanced it may be restored by slowing the pace of the work until more oxygen becomes available, by diffusion of the unbalanced ions into the blood stream and by their excretion through the kidneys and lungs.

Fatigue is a term applied to a state resulting from work. Up to the present there has been no entirely satisfactory definition formulated. It has been divided into mental and physical fatigue. The symptoms arising from mental exertion are relegated to the psychologist[4] and those from physical exertion are allotted to the physiologist. Since fatigue is considered an important limiting factor of work one is forced to accept some definition of it if he plans to investigate the factors governing production. Dill[5] has a most stimulating approach to a classification of the types of fatigue. He has pointed out that moderate work which requires an energy expenditure of three times the basal metabolic rate does not induce exhaustive changes in the muscle. The seat of fatigue due to work of this magnitude must be found elsewhere than in the muscle. He classified these symptoms under the term of *boredom*. Perhaps we might place in this class also *loss of motivation*. The amount of muscular energy transformed is small. The effects of the physical environment on changes in the muscle is almost negligible. The seat of such symptoms should be found in the cerebrum, in the synaps of the nervous pathways leading to the muscle, or in purely psychological attitudes. When hard work is performed, which requires an expenditure of energy eight times the basal metabolism, then fatigue changes occur within the muscle. With further expenditure of energy (maximal work) the continuation of work becomes intolerable.

Environmental conditions now become most important. A significant increase in the heat load, or a change in the conditions which make heat elimination more difficult may cause a complete stoppage of work. Under these conditions the circulation is unable to deliver sufficient blood and oxygen to maintain the normal physico-chemical reactions necessary for muscular contraction. The seat of this fatigue is to be attributed to the failure of the general service functions of the body, circulation, urinary secretion, and respiration.

This is a most helpful outline to follow if one attempts to compare performance of groups of workers under experimental conditions. Even if one is unable to secure quantitative measurements of the fatigue in these various fields, it is most important to locate the field in which performance lags.

We may now return to the original question which provoked this discussion. If one is to compare the effects on production of a comfortable with an uncomfort able environment, he can select the field in which he wishes to develop fatigue symptoms by the severity of the assigned work. It will be necessary next to develop a quantitative factor for the physiologic stress that each environment imposes. This question will doubtless have to be solved by obtaining some integrated factor for the additions and subtractions of heat by a given non-uniform environment to and from a human subject.

Dr. Belding and Professor Hatch[6] presented before the Society an index for evaluating heat stress developed in certain industrial situations where the stress is of sufficient magnitude to cause physiologic strains. If this or some modification of it could be developed for varying conditions where the stress is less severe, then it should be possible to measure the effect on production of any selected environmental condition.

I should like to close by pointing out that productive and successful laboratory investigations in the fields, which I have discussed, require research teams composed of men trained and experienced in both the physical and biological sciences, and elaborate physical facilities for controlling and evaluating the environments and for measuring the responses of human beings to the different environments.

Emphasis should be placed on fundamental studies which are time consuming. Laboratory studies should be supplemented with controlled field investigation.

REFERENCES

1. Physiological Responses to Heat, by H. C. Bazett (*Physiological Reviews VII*, p. 531).

2. (*a*) ASHVE RESEARCH REPORT No. 1320—Physiological Adjustments of Human Beings to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, S. E. Telser, R. W. Keeton, F. K. Hick and M. K. Fahnestock (ASHVE TRANSACTIONS, Vol. 53, 1947, p. 327). (*b*) ASHVE Research Report No. 1342—Comparison of Physiological Adjustments of Human Beings During Summer and Winter, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton and M. K. Fahnestock (ASHVE TRANSACTIONS, Vol. 54, 1948, p. 307). (*c*) ASHVE RESEARCH REPORT No. 1350—Physiologic Adjustments of Normal Subjects and Cardiac Patients to Sudden Change in Environment, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton, I. R. Callen, F. K. Hick and M. K. Fahnestock (ASHVE TRANSACTIONS, Vol. 55, 1949, p. 27). (*d*) ASHVE RESEARCH REPORT No. 1380—Physiologic Examination of the Effective Temperature Index, by Nathaniel Glickman, Tohru Inouye, R. W. Keeton and M. K. Fahnestock (ASHVE TRANSACTIONS, Vol. 56, 1950, p. 51.) (*e*) ASHVE RESEARCH REPORT No. 1447—Physiological Adjustments of Clothed Human Beings to Sudden Change in Environment—First Hot Moist and Later Comfortable Conditions, by F. K. Hick, Tohru Inouye, R. W. Keeton, Nathaniel Glickman, and M. K. Fahnestock (ASHVE TRANSACTIONS, Vol. 58, 1952, p. 189). (*f*) ASHVE RESEARCH REPORT No. 1463—A Comparison of Physiological Adjustments of Clothed Women and Men to Sudden Changes in Environment, by Tohru Inouye, F. K. Hick, R. W. Keeton, J. Losch and Nathaniel Glickman (ASHVE TRANSACTIONS, Vol. 59, 1953, p. 35). (*g*) ASHVE Research Report No. 1508—Physiological Responses to Sudden Changes in Atmospheric Environment—Studies of Normal Subjects, Obese, Hyperthyroid and Hypothyroid Patients, by Tohru Inouye, F. K. Hick, R. W. Keeton and Lionel Bernstein (ASHVE TRANSACTIONS, Vol. 60, 1954, p. 315).

3. The Revolution in Muscle Physiology, by A. V. Hill (*Physiological Reviews XII*, 1932, p. 56).

4. Fatigue of Mental Work, by A. G. Brills (*Physiological Reviews XVII*, 1937, p. 436).

5. The Economy of Muscular Exercise, by D. B. Dill (*Physiological Reviews XVI*, 1936, p. 263).

6. Index for Evaluating Heat Stress in Terms of Resulting Physiological Strains, by H. S. Belding and T. F. Hatch (ASHAE TRANSACTIONS, Vol. 62, 1956, p. 213).

1574

No. 1574

SEMI-ANNUAL MEETING, 1956

WASHINGTON, D. C.

With a total attendance of 646, the 1956 Semi-Annual Meeting took place at the Shoreham and Sheraton-Park Hotels in Washington, D. C. It was the first meeting of the Society in the national capital since 1940. Members and guests came from 33 states as well as from Canada, England, India and Sweden. Attendance figures indicate that 349 members, 82 guests, 161 ladies, and 54 children were registered.

The program was most intensive with events scheduled for practically all available hours from the opening session on June 18 until the end of the banquet late on Wednesday evening, June 20. There were 7 sessions during the 3 days, making concurrent meetings necessary at all times except during the first session on Monday. Fourteen technical papers were presented and discussed, while 2 sessions were of the symposium type.

Pres. John W. James, Chicago, Ill., immediately after opening the first session on Monday, June 18, at 9:30 a.m. in the West Ballroom of the Shoreham Hotel and presenting the Society Officers, asked Chairman Achenbach to outline the meeting program. Mr. Achenbach responded by also extending a welcome on behalf of the Washington Chapter.

Following this, President James announced that the Committee on Resolutions would consist of G. A. Linskie, Dallas, Tex., as chairman; D. Lorne Lindsay, Montreal, Canada; and H. G. Gragg, Chicago, Ill.

President James then announced that 2 technical papers were scheduled (see program on page 312) for presentation at this session. Following their presentation and discussion, President James adjourned the session at 11:00 a.m.

First Vice Pres. P. B. Gordon, New York, N. Y., opened the second session in the Main Ballroom of the Shoreham Hotel and announced that L. G. Miller, East Lansing, Mich., scheduled to act as vice chairman during the presentation of the technical papers had found it impossible to be at the session. In Mr. Miller's absence Vice President Gordon presided and 3 technical papers were presented. After the completion of the discussions which followed the reading of each of these technical papers, Vice President Gordon adjourned the session at 12:20 p.m.

Immediately after calling the third session to order in the West Ballroom at the Shoreham Hotel on Tuesday, June 19, at 9:30 a.m., Second Vice Pres. E. R. Queer,

University Park, Pa., announced that R. S. Dill, Washington, D. C., would act as vice chairman during the presentation of the 3 technical papers scheduled for this session.

Vice Chairman Dill called for discussion of each of these papers following its presentation. On the completion of this discussion period, Mr. Dill turned the gavel back to Vice President Queer who adjourned the session at 11:40 a.m.

President James presided in the Main Ballroom of the Shoreham Hotel at the opening of the fourth session on Wednesday, June 20, at 9:30 a.m., and at once introduced W. F. Friend, New York, N. Y., who presided as vice chairman during the presentation of the 3 technical papers. Following the discussion period, Vice Chairman Friend announced that President James would again take charge of the session which he adjourned at 11:40 a.m.

The fifth session which was organized in symposium form was called to order by First Vice Pres. P. B. Gordon in the West Ballroom at the Shoreham Hotel on Wednesday, June 20, at 9:30 a.m. He at once introduced T. H. Urdahl, Washington, D. C.

The theme of the symposium was the position that a national Society should take in the problem of Standards and Codes as related to its particular activities.

C. H. Pesterfield, East Lansing, Mich., the moderator had arranged for speakers to be present and state their views on 6 topical subdivisions of this general theme as listed in the program. In the absence of Mr. Hunter, his prepared material was presented by M. W. McRae, directing engineer, Heating Engineering Dept., Crane Co., Chicago, Ill.

Professor Pesterfield, acting as moderator for the Symposium, then called on each of these speakers. Following completion of the formally prepared papers, the session was open for questions and comments from the floor.

That those present had found the subject matter of interest was shown by the discussions which were at once brought forth. Thirteen persons took part in the discussions and questions.

Abstracts from the remarks of the speakers at this Symposium session can be found on pages 138 to 143 of the August, 1956 issue of the JOURNAL. There also, there is information concerning plans for making available a substantially full record of the proceedings at the session, including both the speakers' papers and the discussions.

Following the close of the question period, Vice Chairman Urdahl turned the meeting back to First Vice President Gordon who adjourned the session at 12:05 p.m.

Meeting in the Main Ballroom at the Shoreham Hotel on Wednesday, June 20, at 2:00 p.m., the sixth session was called to order by Second Vice Pres. E. R. Queer. He announced that W. A. Grant, Syracuse, N. Y., would take charge of the session as vice chairman. Mr. Grant then outlined the nature of the session which had been organized in the form of a symposium covering the general subject of year round Residential Air Conditioning. Since L. H. Haeger, Levittown, Pa., had found it impossible to attend the session, Mr. Grant acted as moderator in his place. Five topical papers on this general subject had been prepared as listed in the program.

Each of these speakers was introduced in turn by Moderator Grant and each read his prepared paper. Following completion of the papers Mr. Grant then indicated that the session would be concluded by affording those present an oppor-

tunity to express their views concerning the general subject or to ask questions concerning the remarks of the speakers. Eight people took part in the discussion period which followed.

A greatly shortened version of the prepared papers of the several speakers appears in the JOURNAL SECTION, *Heating, Piping and Air Conditioning*, on pages 132 to 139 of the September, 1956 issue. There, too, will be found directions for obtaining a fuller record of this session in Bulletin form.

Following completion of the discussion period, Pres. John W. James took charge of the session and called on G. A. Linskie who presented the Report of the Resolutions Committee.

WHEREAS, the 62nd Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS is now drawing to its close, and

WHEREAS, this wonderful meeting held in our nation's capital, the depository for our privileged tax payers, and where the marble halls house the magnificent memories of the past and the delicate destiny of the future, and

WHEREAS, the technical life of the Society has been advanced through the presentation and discussion of a series of well prepared papers and symposia, and

WHEREAS, through the excellent facilities provided, the technical and other committees of the Society have been able to conduct their meetings and accomplish their aims efficiently and comfortably, and

WHEREAS, the Washington Chapter, through its Committee on Arrangements of which General W. A. Danielson is Honorary Chairman, P. R. Achenbach, General Chairman, W. C. Reamy, Vice Chairman, and J. H. Broome, Chapter President as well as the members and ladies have ably provided an excellent program of entertainment of diversified nature for the enjoyment of their guests and providing visitors with an educational tour of national and international interest, and

WHEREAS, the Chapter provided an outstanding and unique children's program, and

WHEREAS, those who were able (and not under the table) enjoyed meeting "Mabel"

BE IT RESOLVED, that sincere thanks and appreciation be extended

TO the entire Committee on Arrangements and to the officers and members of the Washington Chapter,

TO the ladies of the Chapter for their gracious hospitality in entertaining and assisting the visiting ladies and children,

TO Brigadier General Thomas A. Lane for an interesting welcoming luncheon talk, and

TO the authors and discussers of the technical papers and symposia, and

TO the Shoreham Hotel management and its staff for courteous and efficient service, and

TO all members of the Headquarters Staff under the direction of A. V. Hutchinson whose untiring effort is always appreciated and

TO President John W. James, the Officers and Committees of the Society for their fine contributions in sustaining the continued growth and development of our Society in this, our 62nd year.

Respectfully submitted,

GEO. A. LINSKIE, *Chairman*
D. LORNE LINDSAY
HARRY G. GRAGG

After unanimous adoption of this Report and there being no further business to come before the session, President James declared the meeting adjourned at 4:45 p.m.

Pres. John W. James presided over the seventh session in the West Ballroom at the Shoreham Hotel on Wednesday, June 20, at 2:00 p.m., and announced that First Vice President Gordon would act as vice chairman during the presentation of the 3 technical papers programmed for the session. Following the completion of the discussion period, Mr. Gordon adjourned the session.

PROGRAM—SEMI-ANNUAL MEETING

The Shoreham Hotel, Washington, D. C.

June 18–20, 1956

Friday—June 15

8:00 p.m. Executive Committee (*North Room*)
P. B. Gordon, *Chairman*

Saturday—June 16

10:00 a.m. Finance Committee (*North Room*)
A. J. Hess, *Chairman*

1:30 p.m. Regions Central Committee (*South Room*)
E. R. Queer, *Chairman*

Sunday—June 17

9:30 a.m. Council Meeting (*North Room*)

10:00 a.m. REGISTRATION (*Main Lobby*)

10:00 a.m. Research Executive Committee (*Tamerlaine*)
B. H. Jennings, *Chairman*

2:00 p.m. Committee on Research (*South Room*)
B. H. Jennings, *Chairman*

8:00 p.m. MOVIE (*West Ballroom*)
Call Me Madam starring
Ethel Merman, Donald O'Connor, Vera Ellen (Film courtesy of 20th Century-Fox)

Monday—June 18

9:00 a.m. REGISTRATION (*Main Lobby*)

9:30 a.m. FIRST SESSION (*West Ballroom*)
Call to Order—Pres. John W. James
Greetings by P. R. Achenbach, Chairman Committee on Arrangements
Comparative Performance of Indirect Water Heaters, by L. L. Hill, Minneapolis, Minn., and W. S. Harris, Urbana, Ill., presented by Mr. Hill
Natural Convection and Radiation In A Panel-Heated Room, by T. C. Min, L. F. Schutrum, G. V. Parmelee and John Vouris, Cleveland, O., presented by Mr. Min

Monday—June 18 (*continued*)

9:30 a.m. SIGHTSEEING TRIP to Federal Bureau of Investigation (*children welcome*)

12:00 p.m. WELCOME LUNCHEON (*Blue Room*)

Toastmaster: Brig. Gen. W. A. Danielson
Speaker: Brig. Gen. Thomas A. Lane, Engineer Commissioner, District of Columbia

1:00 p.m. GOLF TOURNAMENT (*Congressional Country Club*)

1:00 p.m. INSPECTION Trip—David Taylor Model Basin

2:15 p.m. TAC on Evaporative Cooling (*North Room*)
Stuart Giles, *Chairman*

2:15 p.m. TAC on Heating and Air Conditioning Loads (*South Room*)
H. T. Gilkey, *Chairman*

2:15 p.m. TAC on Thermal Circuits (*Green Room*)
S. F. Gilman, *Chairman*

2:15 p.m. TAC on Weather Data (*Louis Seize Room*)
John Everetts, Jr., *Chairman*

2:15 p.m. TAC on Heat Pump (*Lower West Room*)
F. R. Ellenberger, *Chairman*

2:30 p.m. LADIES' SIGHTSEEING TRIP to Pakistan Embassy and Islamic Center

7:30 p.m. GET-TOGETHER PARTY (*Shoreham Terrace*)

Tuesday—June 19

8:00 a.m. Nuclear Energy Engineering Committee
R. A. Sherman, *Chairman*

9:00 a.m. REGISTRATION (*Main Lobby*)

9:30 a.m. SECOND SESSION (*Main Ballroom*)

Call to Order—1st Vice Pres. P. B. Gordon
Vice Chairman L. G. Miller

Heat Requirements of Snow Melting Systems, by W. P. Chapman and Samuel Katunich, Pittsburgh, Penna., presented by Mr. Chapman

Small-Pipe Perimeter Heating In A Residence, by D. R. Bahnfleth, Urbana, Ill., H. T. Gilkey, Cleveland, O., and C. F. Chen, Providence, R. I., presented by Mr. Bahnfleth

Heat Load Calculation by Thermal Response, by W. R. Brisken, Bloomfield, N. J., and S. G. Reque, Schenectady, N. Y., presented by Mr. Brisken

9:30 a.m. THIRD SESSION (*West Ballroom*)

Call to Order—2nd Vice Pres. E. R. Queer
Vice Chairman R. S. Dill

Air Velocities In Two Parallel Ventilating Jets, by Alfred Koestel and J. B. Austin, Jr., Cleveland, O., presented by Mr. Austin

Dependence of Water Vapor Permeability On Temperature and Humidity, by S. C. Chang, Saskatoon, Sask., and N. B. Hutcheon, Ottawa, Ont., Canada, presented by Mr. Hutcheon

Effect of Unbalanced Air Pressure on Permeance, by F. A. Joy and D. R. Fairbanks, University Park, Penna., presented by Mr. Joy

Tuesday—June 19 (*continued*)

9:30 a.m. LADIES' VISIT TO WHITE HOUSE AND CAPITOL

12:30 p.m. INSPECTION TRIPS TO (1) Naval Ordnance Laboratory* and (2) National Institute of Health*
(* Cafeteria Service Available on Arrival)

1:00 p.m. SIGHTSEEING TRIP to Mt. Vernon (*Boat and Bus*)

1:30 p.m. Guide Committee (*North Room*)
Albert Giannini, *Chairman*

2:00 p.m. Nominating Committee (*South Room*)
J. Donald Kroeker, *Chairman*

7:45 p.m. TAC on Air Cleaning (*North Room*)
E. F. Snyder, Jr., *Chairman*

7:45 p.m. TAC on Air Distribution (*South Room*)
W. O. Huebner, *Chairman*

7:45 p.m. TAC on Solar Energy Utilization
(*Main Ballroom Foyer A*)
R. C. Jordan, *Chairman*

7:45 p.m. TAC on Sorption (*Green Room*)
G. L. Simpson, *Chairman*

7:45 p.m. TAC on Sensations of Comfort (*Lower West Room*)
C. S. Leopold, *Chairman*

7:45 p.m. TAC on Hot Water and Steam Heating
(*Main Ballroom Foyer B*)
H. A. Lockhart, *Chairman*

8:00 p.m. Chapter Publicity Chairmen and Publication Editors (*Tamerlaine*)
J. H. Fox, *Chairman*

Wednesday—June 20

9:00 a.m. REGISTRATION (*Main Lobby*)

9:30 a.m. FOURTH SESSION (*West Ballroom*)

Call to Order—Pres. John W. James
Vice Chairman W. F. Friend
Laboratory For Solar Energy Study at Minnesota, by R. C. Jordan and J. L. Threlkeld, Minneapolis, Minn., presented by Mr. Jordan
Intermittent Ground Grids For Heat Pumps, by G. S. Smith, Seattle Wash., presented by Mr. Smith
Intermittent Heating and Cooling of Buildings, by P. L. Pfennigwerth and Merl Baker, Lexington, Ky., presented by Mr. Baker

9:30 a.m. FIFTH SESSION (*Main Ballroom*)

Call to Order—1st Vice Pres. P. B. Gordon
Vice Chairman T. H. Urdahl

STANDARDS AND CODES SYMPOSIUM
C. H. Pesterfield, Moderator
Standards and Codes in General—J. C. Fitts

Wednesday—June 20 (*continued*)

9:30 a.m. FIFTH SESSION (*continued*)
STANDARDS AND CODES SYMPOSIUM (*continued*)
Protective Standards—Cyril Ainsworth
Governmental Aspects—R. S. Dill
Local and Municipal Standards and Codes—John W. James
Commercial-Industrial Standards—G. S. Jones
Establishment of Standards and Codes through Experimentation—L. N. Hunter

12:30 p.m. LADIES' LUNCHEON AND FASHION SHOW (*Colonial Dining Room*, Sheraton-Park Hotel)

2:00 p.m. SIXTH SESSION (*Main Ballroom*)

Call to Order—2nd Vice Pres. E. R. Queer
Vice Chairman W. A. Grant

SYMPOSIUM ON RESIDENTIAL AIR CONDITIONING

L. H. Haeger, Moderator
New Concepts of Comfort Cooling—W. H. Scott
Combination Fuel Operated Systems—W. E. Hood
Fuel Operated All-Year Systems—A. B. Newton
Electrically Operated Systems—E. R. Ambrose
Future System Developments—W. F. Friend

2:00 p.m. SEVENTH SESSION (*West Ballroom*)

Call to Order—Pres. John W. James
Vice Chairman P. B. Gordon
Performance of One-Row Tube Coils With Thin-Plate Fins, Low Velocity Forced Convection, by D. G. Shepherd
Temperature and Humidity Effect on Odor Perception, by W. F. Kerka and C. M. Humphreys
The Indirect Evaporative Air Cooler, by J. R. Watt and R. A. Bacon
Report of Resolutions Committee
Unfinished Business
New Business
Adjournment

6:15 p.m. CHILDREN'S PARTY AND BANQUET (*Park Room*)

6:30 p.m. SOCIAL HOUR (*West Ballroom*)

7:00 p.m. SEMI-ANNUAL BANQUET (*Terrace Ballroom*)

Speaker: Hon. Arthur S. Flemming Director, Office of Defense Mobilization

Subject: Defense Through Deterrents

Presentation of Honorary Membership Certificates by Pres. John W. James: Herbert C. Hoover, New York, N. Y.; Dr. Charles F. Kettering, Dayton, O.; Dr. Milton S. Eisenhower, University Park, Pa.

Presentation of Past President's Memory Book to
John E. Haines
Award of Golf Trophies
Past President's Research Bowl
Eichberg Memorial Cup
Paul Bunyan Cup

10:00 p.m. Dancing (*West Ballroom*)

COMMITTEE ON ARRANGEMENTS

W. A. Danielson, *Honorary Chairman* P. R. Achenbach, *General Chairman*

W. C. Reamy, *Vice Chairman*

Reception Committee: J. G. Muirheid, *Chairman;* W. J. Caldwell, R. S. Dill, G. S. Frankel, R. A. Grant, H. H. Hill, L. G. Humphrey, Jr., W. C. Jones, W. E. Kingswell.

Sessions Committee: G. C. F. Asker, *Chairman;* J. W. Alexander, H. G. Carleton, J. J. Chaconas, Sr., C. W. Coblentz, I. M. Day, L. H. Holder, S. A. Heider, C. P. Humke, J. S. King, P. H. Loughran, R. C. McMann, J. J. Nolan, Jr., K. G. A. Siggelin.

Transportation Committee: S. R. Allen, *Chairman;* A. C. Crawford, L. T. Davis, E. B. Ferguson, G. W. Gault, C. W. Holliday, S. H. Robbins, Jr.

Banquet Committee: F. A. Leser, *Chairman;* A. E. Beitzell, G. W. Campbell, H. G. Carleton, S. P. Eagleton, P. H. Norair, T. H. Urdahl.

Entertainment Committee: J. H. Broome, *Chairman;* T. F. Burke, A. C. Crawford, I. H. Dale, H. E. Grossman, P. B. Holmes, H. W. Hottel, H. W. Rush, G. R. Walz.

Finance Committee: A. S. Gates, Jr., *Chairman;* G. W. Campbell, W. H. Littleford, J. J. Nolan, Jr., W. C. Reamy, Glegge Thomas.

Ladies Committee: Mrs. J. H. Broome, *Chairman;* Mmes. P. R. Achenbach, G. C. F. Asker, A. C. Crawford, H. E. Grossman, P. B. Holmes, H. W. Hottel, John W. James, R. C. McMann, J. G. Muirheid, E. F. O'Connell, W. C. Reamy, W. T. Smith.

Children Committee: J. W. Morgan, *Chairman;* J. A. Chappelear, C. W. Holliday, L. G. Humphrey, Jr., W. V. McCoy, L. D. Pain, R. B. Ris.

Sports Committee: F. E. McArdle, *Chairman;* T. F. Burke, P. B. Holmes, H. W. Hottel, W. E. Kingswell, E. F. O'Connell.

Publicity Committee: B. P. Evans, *Chairman;* I. M. Day, W. H. Hobbes, Jr., F. E. McArdle, L. D. Pain.

1575

No. 1575

COMPARATIVE PERFORMANCE OF INDIRECT WATER HEATERS

By L. L. HILL* AND W. S. HARRIS**, URBANA, ILL.

IN HOMES using steam or hot water as the heating medium, it is common practice to supply hot water for domestic use by an indirect water heater which is, basically, an exchanger for transferring heat from the boiler to the domestic water. The heat transfer surface generally consists of coiled or *U*-shaped copper tubes submerged in the boiler water with domestic water circulating inside the tubes so that the two water circuits are separated and never mix.

Indirect heaters may be classified as to type (storage or tankless), and as to location with reference to the boiler (external or internal). The external heater has the heat transfer coils encased in a steel or cast-iron shell located outside the boiler; it thus requires the addition of suitable piping to enable boiler water to circulate over the coils. The internal heater is screwed into or bolted onto the boiler and extends directly into the boiler proper. A storage type heater requires less coil surface than a tankless heater because a reserve supply of hot water can be stored in the tank against periods of peak demand. A tankless heater, on the other hand, has no storage tank and requires a large heat transfer surface to provide sufficient hot water to handle maximum loads.

Since performance data on indirect water heaters were meager, an extensive research program on this type of equipment was initiated at the University of Illinois in 1942 as a part of the research program sponsored by the *Institute of Boiler and Radiator Manufacturers*. The first tests were made during the summers of 1942 and 1943 and the winter of 1943–44.[1] These tests were run using an internal storage type heater located in the oil-fired boiler and connected by suitable piping to a horizontal, insulated, 30-gal hot water storage tank.

The present paper deals with a later investigation which included tests on an external tankless heater and on both internal and external storage type heaters. All heaters were used with the gas-fired house heating boiler in the I=B=R Re-

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[1] Performance of an Indirect Storage Type Water Heater, by A. P. Kratz and W. S. Harris, (University of Illinois, *Engineering Experiment Station Bulletin* 366, 1947).
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956.

search Home. The two storage type heaters were tested with both 30 and 66 gal horizontal hot water storage tanks.

Basic objectives of the test program were to determine the availability of hot water and the cost of heating water with each of three types of indirect water heaters under both summer and winter conditions of operation.

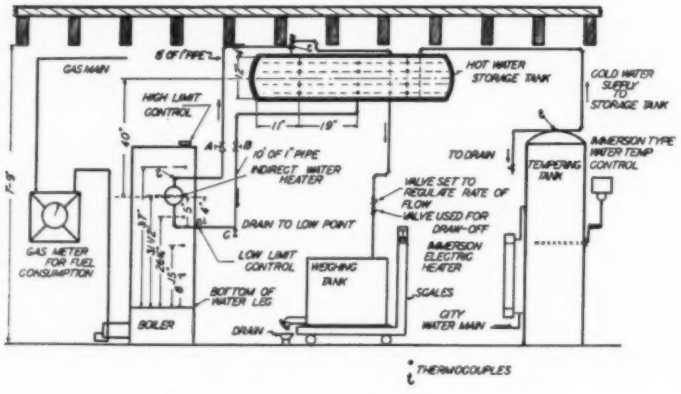


FIG. 1—DIAGRAM OF TEST EQUIPMENT

DESCRIPTION OF EQUIPMENT

The Research Home was a two story dwelling designed and furnished to represent a well-built American home of 1940. Total calculated heat loss for the house (excluding basement) was 43,370 Btuh at temperatures of −10 F outdoors and 70 F indoors.

The domestic water heating system consisted of a gas-fired boiler, an indirect water heater, a water tempering tank, and, in the case of storage type heaters, a hot water storage tank. The boiler was equipped with a conversion-type gas burner, adjusted to a burning rate of approximately 100 cfh. Fuel was natural gas—heating value 1000 Btu per cu ft. A schematic diagram of a typical test setup is shown in Fig. 1.

During the 1948–49 heating season, when winter operation of indirect water heaters was studied, a one-pipe, forced-circulation, hot water heating system was used with convectors in all rooms.

The three heaters used are shown in Fig. 2. The internal storage type heater was of the trombone type consisting of three ¾ in. copper *U* tubes having a heating surface of nearly 2 sq ft. It extended 21 in. into the back section of the boiler and was located 4 in. above the level of the low limit control. Insulated 1-in. pipe connected the heater to the storage tank.

The external storage type heater consisted of a single, helical, copper coil in a cast-iron shell measuring 12½ x 7½ in. Although located outside the boiler, it

was placed at the same level as the internal storage type heater in order to keep the circulating head and the length of pipe connecting the heaters to the storage tank the same in both cases. Like the internal heater, the heating surface of the coil was about 2 sq ft. The external heater was covered with magnesia insulation to a depth of about 1 in. and was connected to the boiler with insulated 1¼-in. pipe and to the storage tank with insulated 1-in. pipe.

The tankless heater was composed of copper *U* shaped coils having a heating surface of 14 sq ft, located within a cast-iron shell 11½ x 16¼ x 8½ in. overall. Boiler water circulated to the heater through insulated 2-in. pipe, and ½-in. pipe was used on the domestic hot water circuit. The heater was covered with 1 in. of magnesia insulation.

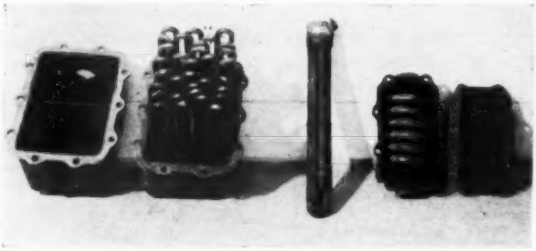


FIG. 2—HEATERS USED IN TESTS

Thirty gal and 66 gal storage tanks were used. The shell of the 30-gal tank, 12 in. in diameter and 60 in. long, was insulated on the sides with 4-ply corrugated asbestos 1 in. thick, and on the heads with 1 in. of magnesia insulation. The 66-gal tank, 18 in. in diameter and 60 in. in length, was similarly insulated but the depth of the insulation was 1½ in. The measured water capacities were 242 lb and 515 lb for the 30-gal and 66-gal tanks, respectively. The tank used to store the hot water was suspended in a horizontal position near the basement ceiling.

Operation of the burner and circulator was regulated by a conventional set of controls consisting of a room thermostat, a low limit control, a high limit control, a magnetic gas valve, and a safety pilot. The heat-anticipating room thermostat, which controlled the operation of both burner and circulator, was placed in the living room 30 in. above the floor. The direct-acting, low limit control, of the immersion type, was located in the rear section of the boiler approximately 26 in. above the bottom of the water leg. This control started the burner whenever the temperature of the water in the boiler reached a prescribed minimum. The high limit control was located in the top of the back section of the boiler. A flow control valve prevented circulation of boiler water through the heating system except when the circulator was operating.

In order to regulate the temperature of the cold water supply the tempering tank was equipped with an immersion type water temperature control operating a 250-watt immersion electric heater.

Copper constantan thermocouples were used to measure the temperatures of: (*a*) the cold water supply, (*b*) the water drawn off, (*c*) the domestic water at the

inlet and outlet of the heater, (*d*) the boiler water entering and leaving the heater, and (*e*) the water at different levels in the boiler and storage tank. These thermocouple locations are shown in Fig. 1. A 10-point recording potentiometer provided a continuous record of water temperatures at selected points in the system. Water was weighed on platform scales sensitive to 0.01 lb, and the gas consumption was measured by a special test meter reading directly to 0.01 cu ft.

AVAILABILITY OF HOT WATER

Automatic Washer Draw Off Tests: Water requirements for an automatic washing machine present one of the heaviest demands for hot water. Consequently water was drawn off at intervals simulating automatic washing machine demands for hot water, both as to quantity and frequency of draw off. The schedule used is given in Table 1. It represents a complete wash cycle from the time clothes are loaded into the machine until they are taken out damp dry. It was developed from hot water requirements as obtained from several automatic washing machine manufacturers. The quantities shown are based on a total water requirement of 8 gal per fill. To test for the availability of hot water from any heater, hot water was drawn at the times and in the quantities indicated in Table 1, and the complete cycle was repeated as often as the water was hot enough for use.

TABLE 1—AUTOMATIC WASHER SCHEDULE

TIME (MIN)	OPERATION	WATER USED (GAL)		WATER (F) TEMP FOR OPERATION
		150 F[a]	70 F	
0– 5	Load	—	—	
5–15	Soak	3.00	5.00	90–110
15–27	Wash	8.00	—	140–160
27–28	Spray rinse	2.00[b]	3.00	90–110
28–35	First rinse	3.00[b]	5.00	90–110
35–41	Second rinse	3.00	5.00	90–110
41–45	Dry	—	—	

[a] Figures this column are water supplied by the heater.
[b] Combined in tests to one draw of 5 gal.

A log of typical automatic washer tests is shown in Fig. 3, with results for both the tankless heater and the storage type heaters. Assuming 125 F to be the lowest useful water temperature, Fig. 3 shows that when using a low limit control setting of 185 F, a cold water supply temperature of 80 F, a 66-gal storage tank, and a storage type heater, as many as seven consecutive tubs of clothes could be washed; if the cold water supply was 60 F the limit was five tubs. Because of the large storage capacity of the 66-gal tank, there was little change in temperature during any draw off. Replacing the 66-gal tank by a 30-gal tank limited the number of consecutive washes to three if the temperature of the cold water supply was 80 F, and to two if the cold water temperature was 60 F. Mixing of the cold and hot water was evident in the tests in which the 30-gal tank was used. During the

first cycle, the water temperature dropped from 187 F at the start to below 150 F at the conclusion. In the second cycle, it dropped to 130 F when using 80 F inlet water and below 125 F when using 60 F water. During the wash period of the second cycle (8 gal of hot water used), the water temperature dropped nearly 15 F. Thus, use of the 66-gal tank not only enabled more tubs of clothes to be washed, but also resulted in more uniform water temperatures throughout the time that the water was being drawn.

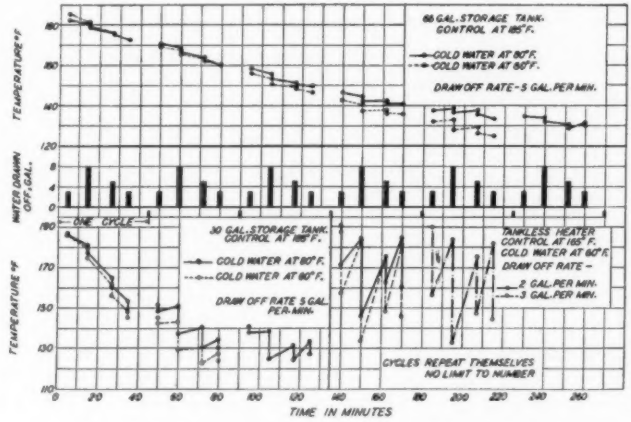

66 GAL. STORAGE TANK. CONTROL AT 185°F.
COLD WATER AT 80°F.
COLD WATER AT 60°F.
DRAW OFF RATE-5 GAL. PER MIN.
TEMPERATURE °F
WATER DRAWN OFF, GAL.
ONE CYCLE
30 GAL. STORAGE TANK. CONTROL AT 185°F.
COLD WATER AT 80°F.
COLD WATER AT 60°F.
DRAW OFF RATE 5 GAL. PER MIN.
TANKLESS HEATER CONTROL AT 165°F.
COLD WATER AT 60°F.
DRAW OFF RATE -
2 GAL. PER MIN.
3 GAL. PER MIN.
CYCLES REPEAT THEMSELVES NO LIMIT TO NUMBER
TIME IN MINUTES

FIG. 3—GRAPHIC LOG—AUTOMATIC WASHER TESTS

There was no limit on the number of wash cycles which could be completed when using the tankless heater as long as the low limit control setting was above about 150 F. However, the variation in water temperature during any draw was appreciable. When the control was set at 165 F and the draw off rate was 3 gpm[2], the water temperature dropped from 185 F to 134 F, a drop of 50 F, between the start and finish of the draw representing the wash period of the cycle. A draw off rate of 2 gpm reduced this temperature drop to 40 F. Using supply water at 80 F rather than at 60 F did not appreciably affect the temperature of water drawn off when the draw off rate was 2 gpm, but at a draw off rate of 3 gpm it reduced the water temperature variation by approximately 5 to 10 F. The drop in temperature experienced with the tankless heaters would have been less had a fast acting water temperature control been used to start the burner as soon as possible after the start of a draw.

A summary of the effect of the control setting, the storage tank size, and the inlet water temperature on the maximum number of tubs of clothes which could be

[2] A 5 gpm draw off rate was used in all tests on storage type heaters to be consistent with standard test procedures used by the *American Gas Association* and the *National Electrical Manufacturers' Association*. With tankless heaters the draw off rate must not exceed the rated capacity of the heater under test hence draw off rates of 2 and 3 gpm were used with these heaters.

washed in succession is shown in Table 2. Since there is difference in opinion as to the minimum water temperature for washing clothes, the number of wash cycles that could be completed before the temperature of the water drawn dropped below both 125 F and 140 F are shown. For a given control setting, and when using 125 F as the minimum usable water temperature, it was possible to complete at least twice as many cycles when using the 66-gal tank as when using the 30-gal tank. However, the superiority of the 66-gal tank over the 30-gal tank was even more apparent when 140 F was taken as the minimum allowable water temperature. Here the mixing action between the hot and cold water that took place in the 30-

TABLE 2—NUMBER OF WASHER CYCLES BEFORE TEMPERATURE OF WATER DRAWN DROPPED BELOW INDICATED MINIMUM

CONTROL SETTING (F)	COLD WATER SUPPLY TEMP (F)	NUMBER OF WASHER CYCLES			
		30-GAL STORAGE TANK		66-GAL STORAGE TANK	
		MIN. TEMP. OF WATER DRAWN		MIN. TEMP. OF WATER DRAWN	
		125 F	140 F	125 F	140 F
185	80	3	1	7	4
	60	2	1	5	3
165	80	2	1	5	3
	60	1	1	4	2
140	80	1	0	no test	no test
	60	1	0	2	1

gal tank limited the number of cycles to one, even at a control setting of 185 F, while with a 66-gal tank as many as four cycles were completed.

WATER HEATING LOADS

Storage Type Heaters: A previous publication[3] contains a discussion of water heating loads imposed on the boiler by indirect storage type heaters. That report indicated a maximum load of approximately 8000 Btuh for water usage of 75 gal per day when using an internal storage heater in conjunction with a 30-gal tank. This load was re-confirmed by the later test series, and, furthermore, it was found that the water heating load was essentially the same for both the external and internal storage heaters and for either the 30-gal or 66-gal storage tank.

Tankless Heater: In the case of the tankless heater, there was no water heating load during periods without hot water demand. However, the moment hot water

[3] Performance of an Indirect Storage Type Water Heater (University of Illinois Engineering *Experiment Station Bulletin* No. 366, pp. 35–37).

was used, the load on the boiler was instantaneous and severe. There had to be sufficient heat available to raise the cold inlet water to the desired hot water temperature during the short period required for the water to circulate through the heater coil. Since the water circulated through the coil at the rate at which it was being drawn from the hot water tap, the load during the draw off sometimes exceeded 150,000 Btuh.

For a tankless water heater to operate suitably, the heat removed from the boiler during any draw off must not exceed the sum of the heat supplied to the boiler by the burner during the draw, the heat stored in the boiler water previous to the draw, and the heat stored in the metal of which the boiler is made. The latter is very small and may be neglected without serious consequence.

The heat stored in the boiler water may be expressed as:

$$H_w = W (t_1 - t_2) \quad . \quad (1)$$

where

H_w = heat stored in boiler water, in Btu.
W = weight of water in boiler, in pounds.
t_1 = temperature of water in boiler at start of draw.
t_2 = minimum temperature of water in boiler during draw.

The heat supplied by the burner would be:

$$H_b = h_f \times E \times F \times T \quad . \quad (2)$$

where

H_b = heat supplied by the burner, in Btu.
h_f = heat content of fuel in Btu per unit used (pounds, gallons, or cubic feet).
E = operating efficiency.
F = fuel burning rate in units used per hour.
T = time burner is in operation, in hours.

The time the burner is in operation during any draw is equal to the total length of the draw off period minus the time delay before the low limit control starts the burner.

The curves of Fig. 4 were computed from the preceding equations, and show that the required heat input to the burner is increased by any one of the following changes: (1) an increase in heater size, (2) an increase in the quantity of water drawn, (3) a decrease in the water content of the boiler.

The curves of Fig. 5, also computed from Equations 1 and 2, show the importance of starting the burner operation as soon after the start of the draw as possible when using a tankless heater. This is especially true for boilers of small water content. The right hand end of each curve represents the maximum time delay that can be permitted regardless of the rate of heat input to the burner if the temperature drop of the water in the boiler is limited to 30 F.

Fig. 5 may be used in selecting the proper combination of boiler, burner, and water heater to meet a given requirement. To illustrate: Determine the minimum firing

rate for the following conditions:

1. Heater size—3 gal. per min.
2. Water content of boiler—25 gal.
3. Burner starts 1 min. after start of draw.
4. Maximum single draw—10 gal.

Curve 10, Fig. 5, shows that the minimum heat input rate should be 77,000 Btu per hr for the stated conditions.

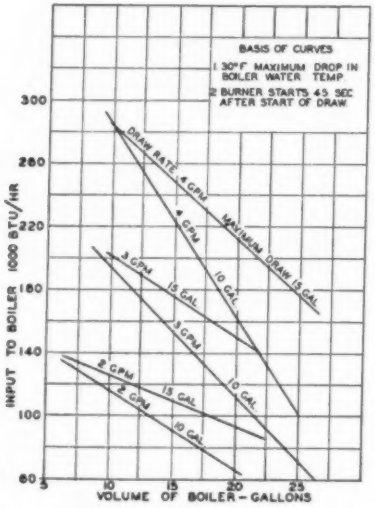


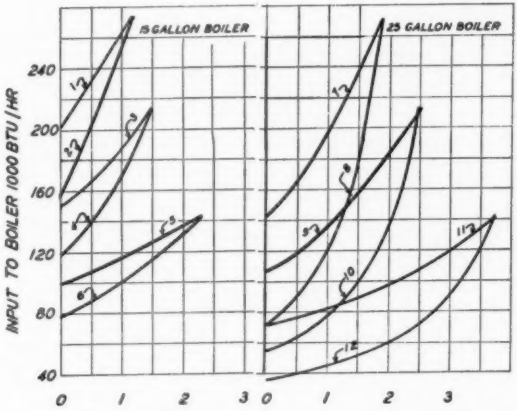
FIG. 4—RELATIONSHIP BETWEEN FIRING RATE, WATER CONTENT OF BOILER, HEATER SIZE, AND MAXIMUM QUANTITY OF WATER DRAWN AT ONE TIME, WITH 70 PERCENT OPERATING EFFICIENCY

Had the water content of the boiler been 15 gal, the minimum heat input rate would have had to be 168,000 Btu per hr (curve 4). Reducing the time delay from one min. to ½ min. would reduce the minimum heat input rates to 63,000 and 138,000 Btu per hr, respectively.

FUEL CONSUMPTION

Winter Operation: During the winter the main load on the boiler was the house heating load, and a large portion of the heat escaping from the surfaces of the boiler, the chimney, the storage tank, and the piping (which was lost heat as far as summer operation was concerned) was used in supplying heat to the house. Accordingly,

several series of tests were run to determine the portion of the winter fuel consumption actually chargeable to heating water under various conditions of operation. Test conditions and resulting data are plotted in Fig. 6. In all cases *DRAW* means *gallons of domestic hot water per day*, and *OUTLET* means *temperature setting of low limit control.*



Curve No.	Draw Rate gpm	Max. Draw gal
1	4	15
2	4	10
3	3	15
4	3	10
5	2	15
6	2	10

Curve No.	Draw Rate gpm	Max. Draw gal
7	4	15
8	4	10
9	3	15
10	3	10
11	2	15
12	2	10

FIG. 5—EFFECT OF TIME DELAY ON MINIMUM FIRING RATE FOR A 15 GAL BOILER AND A 25 GAL BOILER. ALL CURVES ARE BASED ON 30 PERCENT MAXIMUM DROP IN BOILER WATER TEMPERATURE

Since curve 1 fits all test points equally well, it was concluded that the total daily fuel consumption for winter operation was dependent upon the indoor-outdoor temperature difference, but, within the scope of the tests, was independent of the type of water heater used, the size of the storage tank, the setting of the low limit control, the temperature of the cold water, and the quantity of hot water used.

Curve 1 in Fig. 6 gives no clue as to what portion of the daily fuel consumption was chargeable to house heating and what portion to the heating of water. In test Series BA-47 (black diamond symbol in Fig. 6) no water heater was attached to the boiler and the low limit control and flow control valve were not used; therefore, all of the fuel consumption was chargeable to house heating as this was the only function performed by the boiler. Since the curve for Series BA-47 (curve 2, Fig. 6) represents daily fuel consumption for house heating only, and curve 1

represents daily fuel consumption for both house heating and domestic hot water, the difference between these two curves must represent the daily fuel consumption chargeable to water heating for winter operation.

The two curves shown in Fig. 6 differ from one another during mild winter days because at these times the low limit control was dominant over the room thermostat in controlling the boiler water temperature. Any increase in boiler water temperature by action of the low limit control over the water temperature needed to heat

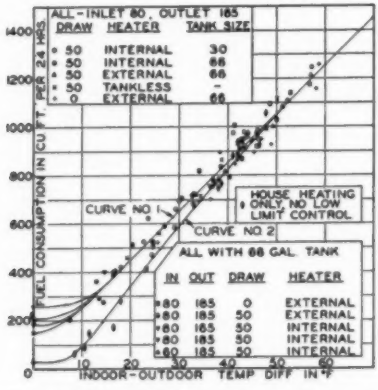


FIG. 6—BURNER PERFORMANCE CURVES FOR HEATING DOMESTIC HOT WATER WITH GAS-FIRED BOILER, WINTER OPERATION

the house meant increased burner operation and increased wasted heat from the boiler and chimney. Where no low limit control was used, as Series BA-47, the water in the boiler was permitted to cool during times when there was no heat demand from the room thermostat. This resulted in a decrease in fuel consumption compared to operation where the boiler water temperature was maintained above a set minimum. As the outdoor temperature got colder, the room thermostat, rather than the low limit control, gradually became the controlling factor in determining the amount of burner operation and the temperature of the water in the boiler. The gas consumption then was identical for all methods of operation.

Summer Operation, Standby: In summer the only function of the boiler was to supply heat for domestic water heating, and, therefore, all fuel consumption was chargeable to the cost of providing hot water. Fig. 7 shows the gas consumption chargeable to the three types of heaters and to the boiler alone for periods during which no hot water was drawn off. The difference between curve 1 and the other curves represents the fuel consumption chargeable to the heat losses of the different heaters, connecting piping, and, when used, the storage tank. With an average boiler water temperature of 180 F at the control level, 15 cu ft of gas per day were required to offset the heat loss from the external tankless heater and piping;

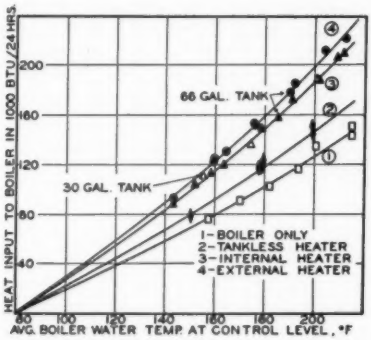


FIG. 7—STANDBY HEAT INPUT CURVES FOR SUMMER (BASEMENT AIR TEMPERATURE AT CEILING APPROXIMATELY 80 F)

46 cu ft for the internal heater, piping, and storage tank; and 56 cu ft for the external heater, piping, and storage tank. The heat losses of the 30- and the 66-gal tanks were so near equal that no significant difference in fuel consumption was noted between them.

For the combinations of boiler and heaters tested, the standby fuel consumption when using the external tankless heater was 22 to 26 percent less than when using either storage type heater, so it is apparent that, even though insulated, the heat loss of the storage tank caused an appreciable increase in the standby fuel consump-

TABLE 3—HOT WATER DRAW OFF SCHEDULES

SCHEDULE	A	B	C
TIME OF DAY	WATER DRAWN OFF, LB		
8:00 a.m.	17	83	83
9:00 a.m.	42	58	83
10:00 a.m.	33	25	100
11:00 a.m.	17	25	42
12:00 noon	17	25	42
2:00 p.m.	17	42	83
5:00 p.m.	17	17	25
7:00 p.m.	17	25	42
8:00 p.m.	25	33	42
9:00 p.m.	50	83	83
Total lb	252	416	625
Total gal	30	50	75

Start of draw off test—at end of burner on-period
End of draw off test—at end of last burner on-period approximately 24 hr after start of test.

tion. Since domestic water heating systems operate at standby a large percentage of the total time, the tankless heater is inherently the more economical to operate and at the same time it releases less unwanted heat into the house during the warm summer months.

Summer Operation, 75-Gal Service Draw Off: The daily fuel requirements for each of the heaters when hot water was used in accordance with schedule C, Table 3, is shown in Fig. 8. The draw off rate when using storage type heaters was 5 gpm, while both 2 and 3 gpm draw off rates were used with the tankless heater.

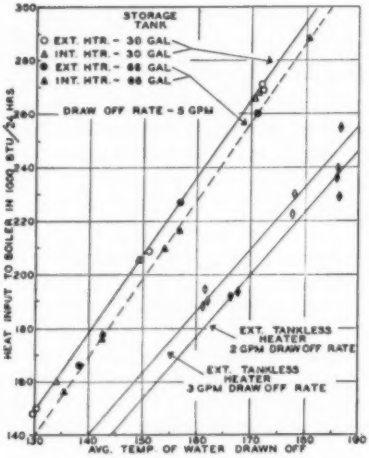


FIG. 8—DAILY HEAT INPUT—STORAGE TYPE AND TANKLESS HEATERS (FOR SUMMER OPERATION WITH 75 GAL OF HOT WATER USED PER DAY, COLD WATER SUPPLY TEMPERATURE 80 F, AND BASEMENT AIR TEMPERATURE AT CEILING APPROXIMATELY 80 F)

Fig. 8 shows that both internal and external storage type heaters required the same heat input for the same storage tank size and operating conditions. However, for a given hot water temperature, the heat requirement of either heater used in conjunction with the 30-gal storage tank was about 8000 Btu per day higher than when used in conjunction with the 66-gal tank. The 30-gal storage capacity was not adequate to provide for a 75-gal service draw off, and, to compensate for the lack of storage capacity, it was necessary to increase the boiler water temperature. This increased the heat losses from the system which, in turn, increased the required heat input. These observations point to the desirability of providing adequate water storage capacity when storage type heaters are used.

The required heat input when using the tankless water heater was 40,000 to 60,000 Btu per day less than that required when using storage type heaters under the same general operating conditions. Over the whole range of water temperatures included in the tests on tankless heaters, reducing the draw off rate from 3 gpm to 2 gpm effected a reduction of about 8000 Btu per day in the heat input.

Fig. 9 shows that reducing the temperature of the cold water supplied to the system from 80 F to 60 F increased the required heat input for the storage type heaters

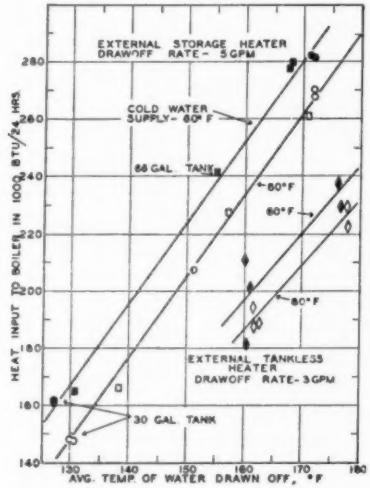


FIG. 9—EFFECT OF COLD WATER SUPPLY TEMPERATURE ON DAILY GAS CONSUMPTION. (CHART BASED ON 75 GAL OF HOT WATER USED PER DAY, AND FOR SUMMER OPERATION WITH BASEMENT AIR TEMPERATURE AT CEILING APPROXIMATELY 80 F)

by 20,000 Btu per day, or about 7 percent, and for the tankless heater by 11,000 Btu per day, or 5.8 percent.

Tests were made to determine the daily fuel consumption when supplying 30 and 50 gal of hot water daily (schedules A and B, Table 3). For brevity, a discussion of these tests is not included in this paper, but will be included in the more detailed report of water heater performance tests now being prepared for publication as a University of Illinois Engineering Experiment Station Bulletin.

Effect of Water Usage on Required Heat Input, Summer Operation: The curves of Fig. 10 were derived from Figs. 7, 8, and 9 and similar fuel consumption data for

water usages of 30 and 50 gal per day. Since, for a given water temperature, the difference in the required heat input for the external and internal storage heaters with either the 30- or 66-gal storage tank was very small, only one set of curves for each water temperature is given. The values of these curves represent average heat inputs, for the two heaters and the two sizes of storage tanks tested.

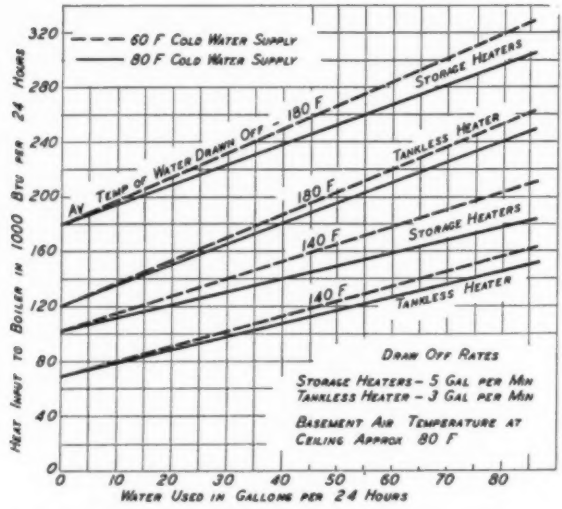


FIG. 10—DAILY GAS CONSUMPTION FOR SUMMER OPERATION. BASED ON SERVICE DRAW OFF TESTS

Fig. 10 shows that for summer operation the tankless heater is much more efficient than the storage type heaters. In fact, the required heat input for the tankless heater was from 20 to 25 percent less than for the storage heaters over the whole range of test conditions. Fig. 10 also shows that the standby losses (no water used) represent approximately 50 to 60 percent of the total heat requirements for summer operation.

WATER HEATING COSTS

Yearly Cost of Heating Water: Because of the number of variables involved, it is not easy to determine the yearly cost of heating water with an indirect water heater. Some of the more important variables are: (1) type and size of heater; (2) average quantity of hot water used per day; (3) temperature of the hot water; (4) unit cost of fuel; and (5) method of control. For the types of heaters and method of control used in the I=B=R Research Home, an estimate of the yearly cost of heating water was made based on the assumptions that the average family of four uses 50

gal of 140 F water per day and cold water is available at a temperature of 60 F. Since it is recognized that operating conditions vary widely, correction factors were determined for other water usage schedules and water temperatures.

TABLE 4—DISTRIBUTION OF AVERAGE OUTDOOR TEMPERATURES AND CORRESPONDING FUEL CONSUMPTION CHARGEABLE TO HEATING WATER

AVERAGE OUTDOOR TEMPERATURE	TOTAL NO. OF DAYS[a]	AVERAGE NO. OF DAYS PER YEAR	GAS REQUIRED TO HEAT 50 GAL OF WATER DAILY (CU FT) WATER DRAWN = 140 F; COLD WATER = 60 F			
			STORAGE HEATER		TANKLESS HEATER	
			PER DAY	TOTAL	PER DAY	TOTAL
−10 to −5	1	0.2	0	0	0	0
−5 to 0	2	0.4	0	0	0	0
0 to 5	4	0.8	0	0	0	0
5 to 10	11	2.2	0	0	0	0
10 to 15	23	4.6	0	0	0	0
15 to 20	38	7.6	0	0	0	0
20 to 25	68	13.6	0	0	0	0
25 to 30	127	25.4	0	0	0	0
30 to 35	169	33.8	20	680	20	680
35 to 40	150	30.0	35	1,050	35	1,050
40 to 45	117	23.4	50	1,170	50	1,170
45 to 50	113	22.6	65	1,470	65	1,470
50 to 55	104	20.8	80	1,660	80	1,660
55 to 60	99	19.8	100	1,980	90	1,780
60 to 65	110	22.0	110	2,420	90	1,980
65 to 70	95	19.0	130	2,470	90	1,710
70 to 75	68	13.6	130	1,770	90	1,220
75 to 80	47	9.4	130	1,220	90	850
80 to 85	17	3.4	130	440	90	300
85 to 90	2	0.4	130	50	90	40
Winter Total		273.0		16,380		13,910
Summer Total		92.0	165	15,180	125	11,500
Yearly Total			(Storage)	31,560 cu ft	(Tankless)	25,410 cu ft

[a] Based on records of United States Weather Bureau Station at the University of Illinois. Includes months of January, February, March, April, May, September, October, November, and December from September 1936 through May 1941.

The frequency of the different outdoor temperatures occurring during the nine months of a typical heating season in Urbana, Illinois, is shown in Table 4. The data are from records of the U. S. Weather Bureau Station at the University of Illinois for the five years from September 1936 to May 1941, inclusive.

Column 4 shows the estimated daily fuel consumption chargeable to heating 50 gal of water daily with the storage type heater for the assumed conditions. The values shown for the winter days represent the difference between the two

curves on Fig. 6 corresponding to indoor-outdoor temperature differences represented by the mean temperature of each bracket shown in column 1. The fuel consumptions shown in column 5 were calculated by multiplying the daily fuel consumptions (column 4) by the average number of days per year having an average temperature falling within the limits of each bracket (column 3). The winter total of column 5 represents an estimated gas consumption of 16,380 cu ft for the storage type heater during the nine months of the heating season.

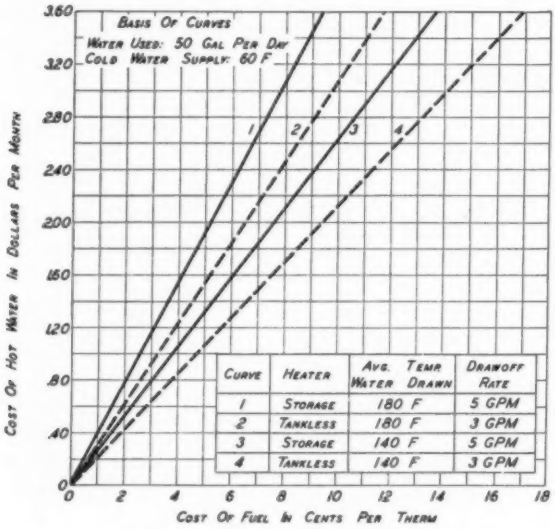


FIG. 11—AVERAGE MONTHLY COST OF HEATING WATER

During the three summer months, a daily fuel consumption of 165 cu ft can be estimated from Fig. 10. Since there are 92 days during the months of June, July and August, the total fuel consumption for the summer months would be 165 × 92 = 15,180 cu ft of gas. The yearly total for the storage type heaters under the conditions shown would be 16,380 + 15,180 = 31,560 cu ft of gas. At a cost of 6 cents per therm (1 therm = 100,000 Btu or 100 cu ft of the gas used in the tests) this estimated fuel consumption gives a total cost of $18.94 per year or an average cost of $1.58 per month.

Similar data are shown for the external tankless heater in columns 6 and 7 of Table 4. The total yearly fuel consumption of the tankless heater is shown as 25,410 cu ft. At a cost of 6 cents per therm, the yearly cost of this quantity of gas would be $15.25, or an average of $1.27 per month.

It will be noted in columns 4 and 6, Table 4, that for days during the heating season with outdoor temperatures high enough that no house heating was required,

the daily fuel consumption charged to heating water was less than during the summer months. This was because the gas consumption of the pilot, amounting to about 35 cu ft per day, was charged to water heating during the summer. Since the principal function of the boiler during the heating season was to supply heat to the house, it seemed logical to charge the pilot fuel consumption to house heating during the heating season. However, had the gas consumption of the pilot been charged to water heating for all days for which the average temperature was 65 F or above, regardless of the month, the additional cost of heating water based on gas at 6 cents per therm would have been only $0.96 per year, or 8 cents per month.

The average monthly cost of heating water as described in the first part of this section was determined for several water usage schedules and hot water tempera-

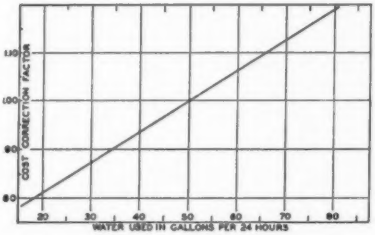


FIG. 12—CORRECTION FACTOR TO OBTAIN COST OF HEATING WATER WHEN USAGE IS OTHER THAN 50 GAL PER 24 HR

tures, and from these data Figs. 11 and 12 were drawn. Curves 1 and 2 of Fig. 11 are for a hot water temperature of 180 F, and curves 3 and 4 represent a hot water temperature of 140 F. Water temperatures between these values may be interpolated linearly between the appropriate curves.

In order to determine a cost correction factor for water usages other than 50 gal per day, the ratio of the fuel consumption for any given water usage and that for a daily water usage of 50 gal was obtained for each combination of heater, storage tank, and hot water temperature included in the test program. It was found that the curve of Fig. 12 represented all conditions within ± 6 percent, and, therefore, could be used to determine the multiplying factor to be applied to the costs obtained from the curves of Fig. 11 to correct them to water usages other than 50 gal per day.

Use of Figs. 11 and 12 can be illustrated by the following example: Determine the average cost of heating water for the following conditions:

Type of heater	External tankless
Hot Water Temperature	180 F
Draw Off Rate	3 gpm
Hot Water Usage	75 gal per day
Fuel	Natural Gas, 1000 Btu per cu ft
Cost of Fuel	10 cents per therm (100,000 Btu)

On Fig. 11 find cost of gas, 10 cents per therm, on horizontal axis, go vertically upward until curve 2 (tankless heater, water temperature 180 F) is intersected, then

horizontally to the left. The average monthly cost of supplying 50 gal of hot water per day is found to be $3.08. Since the quantity of hot water used was not 50 gal per day but 75 gal, Fig. 12 must be used to find the cost correction factor to be applied to the above value. For a daily usage of 75 gal the cost correction factor is 1.16. Therefore, the cost of heating water for the conditions stated is $3.08 × 1.16 = $3.57 per month.

DISCUSSION

R. G. HUEBSCHER, Cleveland, Ohio: Apparently, one of the load criterions was the water consumption of an automatic washer. I might mention that many of the modern day washers have suds saver features in that you must return the water to the laundry tub and it runs back to your washer, in which case all you have to supply for the automatic washer is a little bit of rinse water; and the loads may not be nearly as high as it may appear from slide 2.

K. O. SCHLENTNER, Johnstown, Pa.: This subject is one that has been of great interest to us in the wet heat industry and is a subject on which a great deal of time has been spent during the past years. We are glad to see that this work has been done since it demonstrates the adequacy of this type of water heating.

We are quite sure that several of our government agencies will be keenly interested in the investigation covered by this paper.

One phase of this subject that still merits a great deal of study is that of water demand by families. During the past few years there has been an appreciable increase in the hot water demands in our homes as a result of the introduction of automatic dishwashers and automatic washing machines. We sincerely hope that some way can be found to undertake a study of this kind.

The cost studies were quite revealing since they pointed up the fact that water heaters, used in conjunction with heating boilers, provide a very economical way of providing domestic hot water for a home.

F. F. ROSS*, London, England: In Britain there was an extensive use of small electric storage heaters placed over, or near, sinks which provided sufficient water for intermittent hand washing and washing up. These were very well insulated and had very much lower standby or no-load losses than those indicated in Fig. 10 of the paper.

D. W. NELSON: I have not read this excellent paper in detail and had not expected to comment on it. Some work on storage water heaters at Wisconsin was reported to the Society about 1938†. An important fact found at that time was that the pressure of the water to be heated had an influence on the heating capacity or the heat transfer in the coil. The capacity with water near atmospheric pressure was about double that for pressures above 35 psig.

In searching for the reason it was found that it was due to the formation of steam bubbles on the water side of the heat transfer surface when the water pressure was low, and these bubbles carried the heat into the main body of water thereby increasing the transfer rate. Therefore should a choice of water pressure be available, a low pressure would be preferable.

In my home an indirect water heater and storage tank is used in connection with an oil burning steam boiler. It has been necessary to clean the inside of the helical coil with acid at yearly intervals to maintain reasonable heat transfer. Water is softened by a zeolite softener but mineral deposits have built up on the water side of the coil. A 30-gal storage tank is used and with the use of an automatic clothes washer, the water heating capacity is marginal. The use of a dish washer would call for much more water

* United Kingdom Scientific Mission.
† Paper 1114 Vol 45 Trans.

and at higher temperatures. The oil-burning boiler is operated the year around and the purpose in summer is to heat water in this indirect heater. A gas heater is also available in connection with the same storage tank but has not been used for several years. The water heating capacity with the boiler is greater and moisture problems in the basement are reduced.

In commercial kitchens enormous quantities of water at high temperatures are required for washing and sterilizing of dishes. Heating of wash and rinse water is a very large part of the total heating load and fuel cost. The authors are to be congratulated on an excellent paper that will aid greatly in the solution of water heating problems in such cases as well as in residences.

R. L. MAHER, Waterbury, Conn.: Apparently the coils in these heaters were all plain coils; that is, one hundred percent prime surface. There are coils being used today with fins on the outside.

I would like to ask the authors if they have had any experience in testing finned coils; whether they would comment on the fouling of the external fins, that is, on the boiler side. Apparently, this fouling is important inside and I wonder if the fins will have any marked advantage after a period of service when fouling has taken place.

C. H. FLINK, New York, N. Y.: I would like to hear the authors comment on the temperatures that are now required for hot water service and also on possible changes in the quantity of water required.

It is generally recognized that at or below 140 F which is usual for homes, the corrosion is much less than at higher temperatures. Has the new requirement for automatic washers and dishwashers made it necessary to raise the supply temperature to 180 F?

I would appreciate some further comment on the quantity of water per day that is considered necessary. It is used to be common practice to select a thirty gallon tank for a family of five. A 25 gal heater often was sufficient if the input was 25,000 to 30,000 Btu per hour. Apparently for the same family a forty-gallon tank would now be necessary.

E. R. TESKE, Chicago, Ill.: One of the most interesting parts of this presentation was that referring to the effect of sedimentation on capacity. For some reason the chart does not appear in the printed paper. I would like the author to state what period they used on drawoffs, if the period had an effect on the tests they were conducting as to cost, and also what he might suggest as a drawoff schedule for the ordinary owner of a system of this type.

R. C. CHEWNING, Portland, Ore.: I actually hadn't thought to comment on this paper. I have been kept pretty busy and I hadn't been able to study it ahead of time. All I can say at the moment is that I think it is a very interesting paper and that the work covers a number of interesting angles and factors involved in this matter of heating hot water.

MARTIN CHRISTESEN, Englewood, N. J.: I would like to know whether these were steam or hot water boilers. What was the aquastat setting for the boiler water temperature maintained at in the summer?

D. LORNE LINDSAY, Montreal, Canada: Could I just ask two questions? One was whether the authors could make any comment on the use of a vertical tank as opposed to a horizontal tank; and whether they had any suggestions on the method of connecting the pipe connections into the tank itself, whether the drawoff rate would have any effect on the mixing of the water, the cold water filling the tank, on the supply temperature from the tank.

AUTHOR'S CLOSURE (W. S. Harris): Mr. Maher inquired as to the rapidity of fouling of external fins. The coils in the water heaters used in all of the tests made by the authors and their associates were of plain copper tube with no fins on either the inside

or outside. Not having had experience with indirect water heaters having finned coils, I do not know the effect of fins on the service of the heater.

Mr. Flink has commented on the change taking place in hot water requirements in the home. Reliable data on hot water usage are very meager, however it is apparent that hot water consumption per family is higher today than it was a few years back. Estimates of 30 to 50 gal of hot water per day per family of four have been made. We assumed 50 gal per day as being typical usage.

Except for the automatic dish washer, I do not believe that there has been much change in the required temperature of the water. One hundred forty to 150 F is still hot enough for most uses in the home. Water temperatures as high as 180 F are sometimes used in dish washers. As the water temperature is increased, the rate of scale formation in the heater is increased as is also the cost of operation.

It should be pointed out that the recovery capacity of a storage type indirect water heater is low as compared to that of the average direct fired storage heater and for this reason it is desirable to use somewhat larger storage tanks with the indirect heaters. I would recommend that in no case should the capacity of the storage tank to be used with an indirect heater be less than 40 gal.

Mr. Teske commented on the effects of *sedimentation* on the recovery capacity of the heater. The curve indicating these effects as used in the oral presentation of the paper was not included in the printed paper in order to meet space requirements. For those interested, a more comprehensive discussion of these tests has been published by the University of Illinois.‡ With the water available at Urbana we found that operating the heater for a period of 4 weeks without flushing reduced the recovery capacity by about 27 percent. That reduction in itself did not influence the cost of operation, however it did mean that a longer time was required to reheat the water in the storage tank after hot water was used. If the system was not flushed for a period of 8 weeks, not only was the output of the heater reduced by about 40 percent, but the average water temperature in the storage tank after complete recovery was about 8 F below normal. A 75-gal service draw off test run at this time with a control setting of 185 F resulted in the temperature of the water drawn off being some 15 F cooler than for a comparable test made after flushing the system. In our tests the system was flushed for each test series. In normal useage the heater probably should be flushed about once a month for best results.

In reply to Mr. Christensen, a hot water boiler having a water content of about 15 gal was used in all tests. Several settings of the boiler water temperature control were used to provide different hot water temperatures. The test program included hot water temperatures ranging from a low of about 140 F to a high of about 185 F. The actual boiler water temperatures used ranged from about 160 F to about 200 F. To provide hot water at 165 F in the summer we found that the minimum boiler water temperature had to be maintained at about 165 F when using a tankless heater and at about 185 F when a storage heater was used.

Mr. Lindsay requested comments on the effect of pipe and storage tank arrangements on the performance of storage heaters. This is a phase of the investigation which was not covered in the paper presented at this meeting. These effects were studied however, and the results are included in the University of Illinois publication referred to earlier. On the basis of our tests we would not recommend the use of a horizontal tank if the tank diameter is 12 in. or less. Vertical tanks should be located as high above the heater as possible and in no case should the center of the tank be lowe. than the water heater outlet. Circulating lines between the heater and storage tank should connect to the highest and lowest parts of the tank.

The authors are appreciative of the questions raised by the discussers already referred to in this closure as well as of the comments and suggestions made by Messers. Nelson, Ross, Schlentner, Huebscher and Chewning. The many comments and questions are indicative of the widespread interest in the subject of indirect water heating.

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‡ Performance of Three Types of Indirect Water Heaters, by W. S. Harris and L. L. Hill. (University of Illinois, *Engineering Experiment Station Bulletin* 432).

1576

No. 1576

NATURAL CONVECTION AND RADIATION IN A PANEL-HEATED ROOM

By T. C. MIN*, L. F. SCHUTRUM**, G. V. PARMELEE†, AND J. D. VOURIS‡, CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

CONVECTION data previously available for panel heating design calculations are those obtained from experiments with heated wires, pipes, or small plates with free edges, or with parallel plates only a few inches apart. There has therefore been some uncertainty regarding the application of these data to panel heating. Under the guidance of the Technical Advisory Committee on Panel Heating and Cooling[0], research has been conducted in the ASHAE Environment Laboratory since November 1951 to develop basic information pertaining to panel heating and cooling applications.

In four previous papers[1–4], the total heat flow rates from ceiling panels and floor panels under specific conditions were reported. The influence of various factors affecting these heat flow rates were also reported and discussed. In order to provide a more general approach to design, this paper presents an analysis of the convection and radiation components of the total heat exchange at all the surfaces of a room heated by warm panels. The results of this study, with certain limitations, may be useful in solving panel heating problems of various degrees of complexity. The convection and radiation coefficients can be combined for estimating panel outputs, or they may be used separately as coefficients in rational design procedures

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‡ Assistant Research Engineer, ASHAE Research Laboratory and now with Army Ordnance, Philadelphia, Pa.

[0] P. B. Gordon, Chairman; W. S. Harris, Vice Chairman; A. B. Algren, W. P. Chapman, John Everetts, Jr., H. T. Gilkey, E. S. Howarth, L. N. Hunter, A. T. Jorn, W. C. Kadow, G. D. Lain, H. A. Lockhart, C. O. Mackey, R. L. Maher, S. K. Smith, E. F. Snyder, Jr., W. F. Spiegel, J. M. Van Nieukerken, G. L. Wiggs.

[1] Exponent numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR CONDITIONING ENGINEERS, Washington, June 1956.

such as Hutchinson's, or in the thermal-circuit approach to unsteady-state problems as advanced by Nottage.

Data were obtained mainly from the previous studies but include results from recent experiments. Experiments were made under steady-state conditions with a uniform environment, wherein all room surfaces other than the heated panel were at the same temperature, and with the entire floor or ceiling heated. Other reports on this program will follow.

TEST ROOM AND APPARATUS

The ASHAE Environment Laboratory was described in a previous paper[5]. All 6 surfaces of the room are composed of aluminum panels, the temperatures of which are controlled by liquid circulation. The surfaces are instrumented for direct measurement of heat-flow rates and surface temperatures, and for these tests were painted with a semi-gloss gray paint over 2 coats of primer.

For radiation measurements, a sensitive radiometer[6] with its field of view restricted to narrow cone angles was used.

PROCEDURE

The total exchange between a room surface and its environment is the sum of the convection exchange between the surface and the room air and the radiation exchange between this surface and the other room surfaces which it can *see*. In all cases total heat exchange in the Environment Laboratory was measured by plate-type heat-flow meters[7] distributed over and fastened to the room-side surfaces of the walls, floor, and ceiling. The problem, therefore, was one of separating this total exchange into its convective and radiative components.

The total heat output of the panel surface was measured directly by plate-type heat-flow meters, and the radiation output was subtracted from the total panel output to obtain the convection heat flow. The convection coefficient was then evaluated by dividing the convection heat flow by the temperature differences of the surface and the room air. The radiation exchange was measured with the radiometer at a number of heat-flow meter locations by sighting at the meter surfaces and obtaining the total radiant energy leaving them at a given angle, and then sighting in the opposite direction and measuring the total radiant energy impinging on them. For proper summation of the net radiation exchange, readings were taken at a number of positions over the hemisphere. The heat-flow meters were painted the same as the surface on which they were located for most of the radiation measurement. However, for a few tests some of the meters were covered with highly polished foil.

Tests in the Environment Laboratory covered the full range of panel and unheated surface temperatures encountered in practice. Floor panel temperatures were varied from 75 F to 110 F and ceiling panel temperatures from 90 F to 150 F. During both series of tests, the AUST* was varied from 40 F to 70 F. Tests were also made with 3 different sized rooms, 24½ x 12 x 8 ft; 24½ x 12 x 12 ft; and 12 x 12 x 8 ft.

A few tests were made to determine the convection heat transfer from an 11½ in. square plate in various orientations, with free and with baffled edges.

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* The area weighted average temperature of the unheated surfaces of the room. In a uniform environment, AUST equals the temperature of all surfaces other than the heated panel.

In most of the tests, the relative humidity was low, around 20 percent, and the effect of water vapor absorption and reradiation was not evaluated.

RADIATION

Separation of the convective component from the total heat exchange at a room surface can be accomplished if the net radiant energy exchange can be evaluated. In radiation interchange between any 2 surfaces forming part of an enclosure, there are 2 factors involved besides the well-known Stefan-Boltzmann law for black-body radiations. These are: (1) the view the surfaces have of each other; and (2) their emitting and absorbing characteristics. The former involves a shape factor or view factor which takes into account both the shape and the relative position of the surfaces. Information concerning this has appeared elsewhere.[8–11] The latter so-called emissivity factor or *effective* emissivity depends upon the relative sizes, relative positions, shapes, reflection characteristics (diffuse or specular) and the individual emissivity of the surfaces.

The general equation for radiation interchange given by Hottel[11] is

$$q_r = F_{1-2}\,\sigma(T_1^4 - T_2^4) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

F_{1-2} = the interchange factor involving both items 1 and 2 as just stated.

When the reflections among 6 surfaces of a room are considered along with surface irregularities and the absorbing water vapor in the air, theoretically exact treatment of radiation exchanges is impractical. The procedure, then, was to rely on actual test measurements.

For accuracy in evaluating the radiation exchange, the simplest configuration possible was used, namely, all room surfaces except the heating panel at the same temperature. The average net exchange between the entire surface of the heated panel and the surroundings was determined from net exchange measurements at various locations on the surface. The interchange factor between the floor panel and its environment was found to be 0.876. If the surface is a perfectly diffusing emitter and reflector, *i.e.* obeys the cosine law, and if its emissivity is independent of temperature, an interchange factor F_{1-2} can be found from the following equation given by Hottel[11]

$$F_{1-2} = \frac{1}{\frac{1}{\bar{F}_{1-2}} + \left(\frac{1}{e_1} - 1\right) + \frac{A_1}{A_2}\left(\frac{1}{e_2} - 1\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The angular emissivity of the painted surfaces in the Environment Laboratory was measured by Umur, Parmelee and Schutrum[6] for incident angles up to 85 angular deg. This same report gave the hemispherical emissivity of the paint as 0.88 and normal emissivity of 0.92.

For the conditions existing in the test room average $\bar{F}_{1-2}$ is unity for the floor panel and its environment; e_1 and e_2 = 0.88; and A_1/A_2 = 300.9/884.2 = 0.34. Applying these values to Equation 2, F_{1-2} becomes 0.85 which compares favorably with the measured value of 0.876. Furthermore, the hemispherical emissivity of the panel surface, 0.88, is nearly the same as the observed interchange factor, 0.876. Therefore, the interchange factor for a panel in a completely enclosed

space appears to be equivalent to the hemispherical emissivity of the panel surface ($F_{1-2} = e_1$). It is not known whether this is true for surfaces having low emissivities.

Although one would expect the interchange factor for the heated ceiling to be the same as for the heated floor panel, it was found to be 0.90, which is slightly higher than the 0.876 value for the floor. This may be due to the fact that meters in-

NOMENCLATURE

A = area of a surface, square feet, A_1 refers to surface 1 (enclosed body). A_2 refers to surface 2 (enclosing body).

c = dimensionless exponent.

c_p = specific heat of air at constant pressure, Btu per ($pound_m$) (Fahrenheit degree).

D_e = equivalent diameter equals four times area divided by the perimeter of surface, feet.

e = emissivity of surface for radiation, dimensionless; e_1 of surface 1; e_2 of surface 2.

F = overall interchange factor, dimensionless; F_{1-2} from surface 1 to surface 2.

$\overline{F}$ = black-surface overall interchange factor, dimensionless; $\overline{F}_{1-2}$ from surface 1 to surface 2.

g = acceleration due to gravity, 4.17×10^8 ft per (hour)^2.

H = height of walls, feet.

h = coefficient of heat transfer, Btu per (hour) (square foot) (Fahrenheit degree); h_c for natural convection from a surface to ambient air; h_{rc} for combined coefficient for radiation plus convection.

k = thermal conductivity of air, Btu per (hour) (square foot) (Fahrenheit degree per foot).

L = characteristic linear dimension for convection, feet.

N_{Gr} = Grashof number, $\beta\rho^2 g L^3 \Delta t / \mu^2$, dimensionless.

N_{Nu} = Nusselt number, $h_c L/k$, dimensionless.

N_{Pr} = Prandtl number, $c_p \mu/k$, dimensionless.

q = rate of heat transfer, Btu per (hour) (square foot); q_c for natural convection, q_r for radiation.

T = absolute temperature, Rankine degree; t_1, temperature of surface 1; t_2, temperature of surface 2.

X = length of conduction path, feet.

Greek Letters

α = dimensionless constant.

β = coefficient of volumetric expansion of air, reciprocal Rankine degree.

σ = Stefan-Boltzmann constant, 0.173×10^{-8} Btu per (hour) (square foot) (Rankine degree)^4.

Δt = temperature difference between surface and air, Fahrenheit degrees.

μ = viscosity of air, $pounds_m$ per (hour) (foot).

ρ = density of air, $pounds_m$ per cubic foot.

stalled on the ceiling were constructed with embossed aluminum foil instead of smooth foil as used on the floor meters. The added roughness of these surfaces would be expected to increase the radiation output.

The interchange factor for the polished foil covering on some of the meters was found to be 0.07 which is again in good agreement with Hottel's Equation 2, where A_1/A_2 approached zero, and the interchange factor F_{1-2} equals the emissivity of the foil. The normal emissivity of highly polished aluminum foil is around 0.05[12] and the hemispherical emissivity would be about 0.06.

The interchange factor F_{1-2}, once established by measurement, was used thereafter to calculate the radiation exchange between the warm panel and its environ-

ment in the test room. The radiation transfer to the walls and the ceiling, from the heated floor panel, was obtained by distributing the net radiation output of the panel according to the shape factors between the panel and each of the other surfaces. This was possible because the ceiling and the walls were at the same temperature. The radiation exchanges with a heated ceiling were treated in a similar manner but were somewhat higher, since the interchange factor was 0.9 for the ceiling panel and 0.876 for the floor panel. This difference was judged not

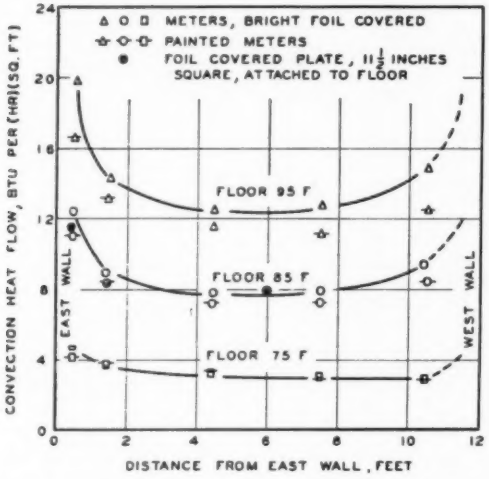

FIG. 1—CONVECTION FROM FLOOR PANEL (UNIFORM ENVIRONMENT, NO INFILTRATION, AUST 65 F, 24½ X 12 X 8 FOOT ROOM)

to be serious. It is interesting to note that the painted surface appeared to behave as a highly specular rather than as a perfectly diffuse reflector (see Appendix). Nevertheless the specular reflections became diffuse after 2 or 3 reflections from other surfaces.

CONVECTION

In a space entirely enclosed by a ceiling, floor, and walls, with no infiltration, natural convection is brought about by differences in density of the air caused by temperature differences. The rate of convection for plane surfaces depends upon the temperature differential and the film coefficient. The film coefficient is related to the mean temperature of the surface and air, temperature differences, size and position of the surface, boundary conditions, and the relative positions of other surfaces.

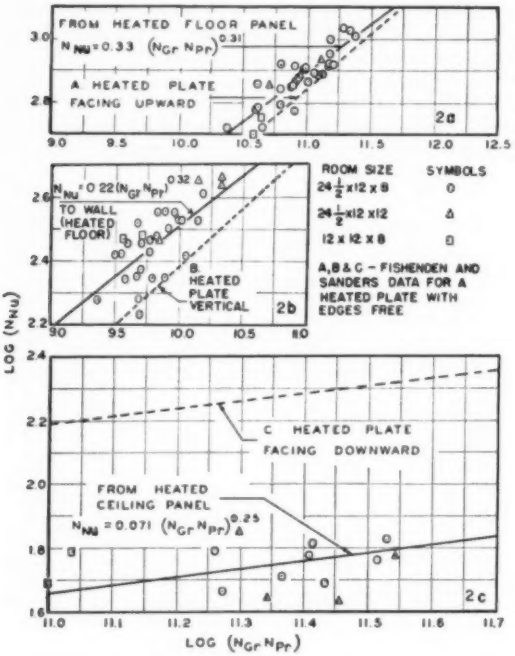
FROM HEATED FLOOR PANEL
$N_{Nu} = 0.33\ (N_{Gr} N_{Pr})^{0.31}$
A HEATED PLATE FACING UPWARD
2a
$N_{Nu} = 0.22 (N_{Gr} N_{Pr})^{0.32}$
TO WALL (HEATED FLOOR)
B HEATED PLATE VERTICAL
2b
ROOM SIZE
SYMBOLS
$24\frac{1}{2}$ x12 x 8
$24\frac{1}{2}$ x12 x12
12 x 12 x 8
A,B & C – FISHENDEN AND SANDERS DATA FOR A HEATED PLATE WITH EDGES FREE
C HEATED PLATE FACING DOWNWARD
FROM HEATED CEILING PANEL
$N_{Nu} = 0.071\ (N_{Gr} N_{Pr})^{0.25}$
2c
LOG (N_{Nu})
LOG ($N_{Gr} N_{Pr}$)

FIG. 2—COMPARISON OF NATURAL CONVECTION FOR ROOM SURFACES AND SIMILARLY ORIENTED SMALL PLATES

For natural convection, the film coefficient or convection coefficient h_c may be expressed in dimensionless terms by Nusselt, Grashof and Prandtl numbers[13] as follows:

$$N_{Nu} = \alpha\ (N_{Gr}\ N_{Pr})^{c} \quad . \quad (3)$$

or

$$h_c L/k = \alpha\ [(c_p \mu/k)(\beta \rho^2\ g\ L^3\ \Delta t/\mu^2)]^{c} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Once the constants α and c are evaluated from experimental data, the equation may be applied to similar problems as long as the shape and orientation of the system remain the same and the product of N_{Gr} and N_{Pr} is within the range of conditions used in the original evaluation.

The physical properties of air, ρ, κ, β, and μ are to be taken at the arithmetic mean of the surface and room air temperatures. In all cases in this report the room air temperature is the temperature at the 5-ft level at the center of the room.

The characteristic linear dimension L is one which relates the size of a surface of given shape and orientation to the convection coefficient. For example, for a long

horizontal cylinder, the diameter affects the convection coefficient greatly, while the length has little effect. In the case of a horizontal square plane surface, the length of a side has been used as the characteristic dimension. For rectangular horizontal plane surfaces, such as the ceiling or floor of a panel-heated space, the characteristic dimension was taken as the equivalent diameter, which is defined as 4 times the hydraulic radius or 4 times the area divided by the perimeter. For vertical plane surfaces, such as walls, the height was the characteristic dimension used.

Correlation of Convection Data, Floor Panel Heating: Convection heat flow was obtained by subtracting the calculated radiation output from the total panel output as described before. The convection from the heated floor was found to be greater near the walls than in the center of the room, as shown in Fig. 1. This was due to the fact that the air near the walls was cooler and at a higher velocity than that at the center of the room. The average area weighted heat flow was, therefore, used for correlation.

The dimensionless groups for 34 tests were evaluated‡ with the equivalent diameter as the characteristic dimension, and correlated by the method of least squares. The analysis indicated that the convection from a heated floor may be represented by:

$$N_{Nu} = 0.33\ (N_{Gr}\ N_{Pr})^{0.31} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Convection from Heated Floor

The test data for the 3 room sizes are shown in the top curve Fig. 2a.

As the temperature range occurring in panel heating systems is small, the physical properties of air may be considered as constants. For practical computation, therefore, the heat transfer coefficient for natural convection may be correlated for a given orientation by means of 2 variables, *viz*: the temperature difference between the heat-transfer surface and the room air, and the characteristic dimension. Equation 5 may therefore be simplified to

$$h_c = 0.39\ [(\Delta t)^{0.31}/D_e^{0.08}] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Convection from Heated Floor

Equation 6 has been plotted in Fig. 3 for a room size 24½ x 12 x 8 ft (D_e = 16.3) and is compared with test data from that room. Curves of other investigators and additional test data for a free-edge small plate are also shown in the figure and will be discussed later.

The convection data for the 4 walls were rather scattered, especially for the north wall where a door is located. From the average of values for the other 3 walls, the convection coefficients were obtained from 25 tests and are expressed in dimensionless form as:

$$N_{Nu} = 0.22\ (N_{Gr}\ N_{Pr})^{0.32} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

Convection to Walls, Heated Floor

The data, in form of Nusselt number and the product of Grashof and Prandtl numbers are shown in Fig. 2b.

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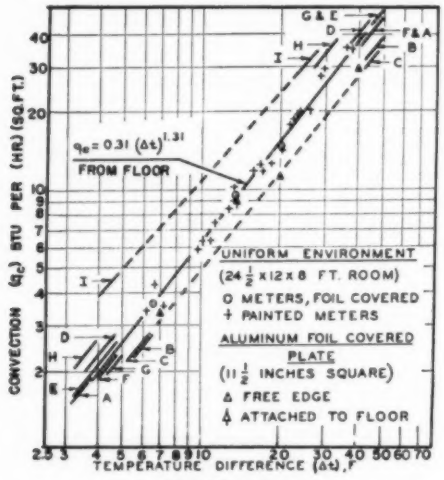
‡ All air properties were taken from *Thermodynamic Properties of Air*, by Joseph H. Keenan and Joseph Kaye (John Wiley & Sons, Inc., New York, 1945) and the HEATING, VENTILATING, AIR-CONDITIONING GUIDE 1955 (published by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York).

Equation 7 may be simplified to

$$h_c = 0.29\ [(\Delta t)^{0.22}/H^{0.05}] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Convection to Walls, Heated Floor

The test data for one room size are shown in Fig. 4.



Notes for Fig. 3

A. Fishenden & Saunders—Turbulent Region Ref. 13.
B. Fishenden & Saunders—Turbulent Region (McAdams) Ref. 11.
C. Fishenden & Saunders—Streamline Region (L = 0.96 Ft) Ref. 13.
C'. Fishenden & Saunders—Streamline Region (L = De = 16.3 Ft) Ref. 13.
D. Griffiths & Davis Ref. 14.
E. Griffiths & Davis (McAdams) Ref. 20.
F. Heilman (Carroll) Ref. 19.
G. King Ref. 16.
H. Nusselt, Henky & Hottinger (Giesecke) Ref. 18.
I. Wilkes & Peterson Ref. 17.

FIG. 3—HEAT TRANSFER FROM HEATED FLOOR PANEL BY NATURAL CONVECTION

The convection to the ceiling when the floor is heated is shown in Fig. 5 and the convection coefficient may be expressed by

$$h_c = [0.58\ (\Delta t)^{0.28}/D_e^{0.16}] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Convection to Ceiling, Heated Floor

Correlation of Convection Data, Ceiling Panel Heating: Since the convection heat flow from a heated ceiling was found to be relatively small, slight errors in radiation measurements resulted in large errors in convection data. To reduce these errors, some of the heat-flow meters were covered with polished aluminum foil, thereby reducing the radiant heat flow and making the convection component a greater percentage of the total.

Convection from the ceiling was also determined directly by measuring the temperature gradient through the boundary air film and computing the heat conducted through this film. The temperature gradient was measured by a fine thermocouple, the position of which could be precisely determined. The device used for measuring this temperature gradient is shown in Fig. 6. The ceiling diffusers and fluorescent lighting shown in the figure were installed after the measurements reported in this paper were made. Typical experimental data obtained by

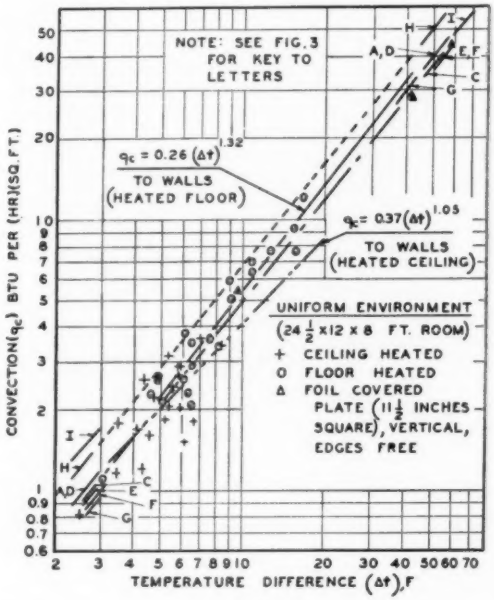

NOTE: SEE FIG. 3 FOR KEY TO LETTERS
$q_c = 0.26\,(\Delta t)^{1.32}$
TO WALLS (HEATED FLOOR)
$q_c = 0.37(\Delta t)^{1.05}$
TO WALLS (HEATED CEILING)
UNIFORM ENVIRONMENT
(24 1/2 x 12 x 8 FT. ROOM)
+ CEILING HEATED
O FLOOR HEATED
Δ FOIL COVERED PLATE (11 1/2 INCHES SQUARE), VERTICAL, EDGES FREE
CONVECTION (q_c) BTU PER (HR)(SQ. FT.)
TEMPERATURE DIFFERENCE (Δt), F

FIG. 4—HEAT TRANSFER TO WALLS BY NATURAL CONVECTION

this method are shown in Fig. 7. The conduction or the equivalent convection through this layer was then computed by the equation

$$q = -kA\,(dt/dX) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

where

k = the conductivity of the air.

dt/dX = the temperature gradient, Fahrenheit degree per foot.

Investigation of the boundary layer thickness was not attempted; however, in one test, measurements were made as far as 0.75 in. from the ceiling and the gradient

was still linear. Although most measurements were steady, variation in thermocouple output increased in amplitude and frequency with increasing distance from the ceiling.

Since both the difference method and the direct method were judged to be equally accurate, data obtained by both methods were correlated. In dimensionless form, the correlation was found to be

$$N_{Nu} = 0.071\ (N_{Gr}\ N_{Pr})^{0.385} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Convection from Heated Ceiling

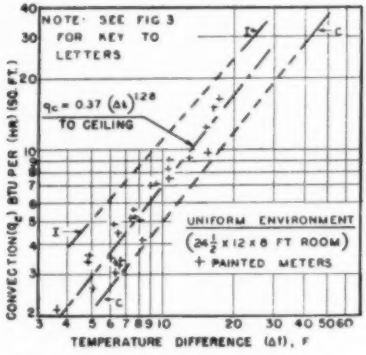


FIG. 5—HEAT TRANSFER TO CEILING BY NATURAL CONVECTION (ROOM HEATED BY FLOOR PANEL)

This relationship is plotted in Fig. 2c, and may be simplified to

$$h_c = 0.041\ [(\Delta t)^{0.28}/D_e^{0.24}] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

Convection from Heated Ceiling

Convection heat transfer for a single room size is shown in Fig. 8. Experimental data determined with painted heat-flow meters are also plotted in Fig. 8 but were not used in determining the equation.

The convection coefficient for the walls was determined from the average test values for 3 walls and may be expressed as

$$h_c = 2.17\ [(\Delta t)^{0.08}/H^{0.80}] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

Convection to Walls, Heated Ceiling

The convection data for one room size are plotted in Fig. 4.

In a ceiling-heated room, the temperature difference between the floor and room air was found to be less than 7 F, and the convection to the floor was less than one Btu per (hour) (square foot). This resulted in considerable scatter of the convection values when determined by subtracting the radiation exchange from the observed total heat flow. Another approach was made by plotting the observed total heat flow to the floor against the difference between the floor temperature

and the area weighted average temperature of all surfaces excluding the floor. Based on the values of the total heat flow read from a smooth curve drawn through these points, the difference method was then used to get the approximate convection.

The convection values were found to lie between the extrapolated curves for free-edge plates and the convection from the heated ceiling shown in Fig. 8. Probably errors accumulated in the determination of the convection heat flow to the floor are of the same order of magnitude as the convection. Hence, no satisfactory equation has been developed. However, no significant error would result if Equation

FIG. 6—INTERIOR VIEW OF TEST ROOM SHOWING THERMOCOUPLE PROBE APPARATUS USED FOR BOUNDARY FILM TEMPERATURE MEASUREMENTS

12, convection from the heated ceiling, were used for calculating the convection to the floor as long as the range of temperature differences encountered in the tests is not exceeded.

DISCUSSION

Convection, Ceiling Heating: The convection coefficient for the ceiling panel was found to be much smaller than that for a small horizontal heated plate facing downward with edges free, as shown in Figs. 2c and 8. Information concerning the latter has appeared in the literature[11, 13–20]. Other investigators[8, 15, 16, 19, 21–23] predicted that the restriction of air currents at the ceiling of a room would result in lower convection rates.

In order to investigate the effect of boundary conditions on the convection from a horizontal surface, and to verify the low measured value of convection from the ceiling, special tests were made with a horizontal free-edged heated plate 11½ in. square facing downward, suspended at the 60-in. level in the center of the room.

The room air temperature was uniform because the floor, ceiling and walls of the room were controlled at one temperature while the small plate was controlled at a higher temperature. The heat flow was measured by a plate-type heat-flow meter cemented to the face of the heated plate. Resulting values of convection agreed well with those of previous investigations as shown in Fig. 8. The edge of the plate was then baffled by a wooden frame 2 in. wide and the convection reduced considerably as shown also in Fig. 8. When the heated plate was attached to and heated to the same temperature as the ceiling, the convection was further reduced. The area weighed value of measurements of several locations is shown in Fig. 8 to lie on the convection curve for the ceiling. In the Environment Laboratory or

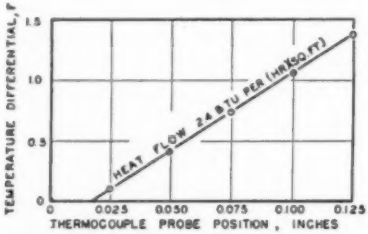


FIG. 7—TEMPERATURE GRADIENTS IN LAMINAR BOUNDARY LAYER ADJACENT TO CEILING PANEL (5½ FEET FROM WEST WALL, 120F CEILING, 65F AUST, NO INFILTRATION, UNIFORM ENVIRONMENT, 24½ X 12 X 8 FOOT ROOM)

any actual room, the ceiling is in a sense baffled by the walls. Convection will be affected by this condition.

Convection, Floor Heating: In a floor-heated room with uniform environment the convection heat transfer from the floor panel to the room air was essentially the same as reported by other investigators for a small horizontal heated plate facing upward. The agreement is shown in Figs. 2a and 3.

Wilkes and Peterson's data, curve I of Fig. 3, are somewhat higher than all the other curves. Their data were obtained with large heated and cooled plates spaced only 4 in. apart. This close spacing may account for the difference.

Tests with the small heated plate were also made in the Environment Laboratory. The 11½ in. square heated plate previously described was faced upward and was located first at the 60-in. level and then at the floor level. The results from these tests are also shown in Fig. 3 and agree well with the other data.

The convection transfer from the air to the cooler ceiling in a space heated by a floor panel is shown in Fig. 5, and agrees well with the data of others. It differs only slightly from the data shown in Fig. 3, for heat transfer from a heated floor to cooler air. This slight difference may be attributed to the fact that the room air temperature on which the temperature differences were based was taken at a point 36 in. away from the ceiling but 60 in. away from the floor. Since the walls were

at the same temperature as the ceiling but at a lower temperature than the floor, the boundary effects may also have contributed.

Convection, Walls: The values of convection to the walls both with floor heating and ceiling heating agree substantially with the test values for a free-edge vertical-plate, as shown in Fig. 4. However the convection to the wall in a ceiling-heated room is a little lower than in a floor-heated room. This may be the result of less air motion in the room or because a higher radiation interchange factor was used when the ceiling is heated. Since all the data for convection to the walls with ceiling heating were taken within a small range of temperature difference, and scattered

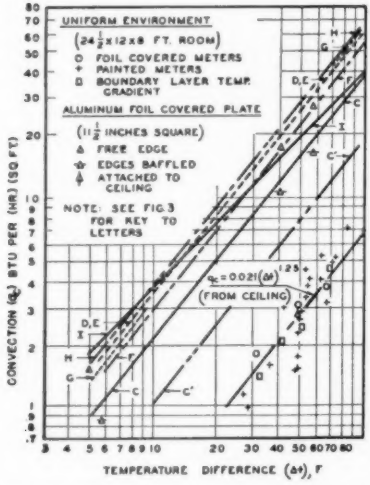


FIG. 8—HEAT TRANSFER FROM HEATED CEILING BY NATURAL CONVECTION

considerably, Equation 13 and the slope of the curve in Fig. 4 may be slightly in error.

General: A summary of the information on test apparatus, test conditions, and convection equations for heated surfaces by various authorities is given in Table 1.

It should be noted that the convection values reported here were obtained in the Environment Laboratory under conditions of no infiltration. In actual panel-heated room, infiltration would cause additional convection. For 1 air change per hour at 0 F in the test room, the convection coefficient increased about 60 percent for the heated ceiling panel at 120 F[1], and increased about 40 percent for the heated floor panel at 85 F[2]. However, the convection coefficients for the walls were not changed appreciably. More definite information will be presented on this subject in another paper.

Characteristic Linear Dimension: Since the data for 3 room sizes correlated well when the equivalent diameter was used in dimensionless Equations 5 and 11 (See

TABLE 1—COMPARISON OF TEST APPARATUS, TEST CONDITIONS, AND NATURAL CONVECTION EQUATIONS FOR PLANE SURFACES

NAME OF INVESTIGATOR	REF. NO.	APPARATUS	SIZE OF TEST SURFACE	AIR TEMPERATURE	TEMP. DIFFERENCE	CONVECTION EQUATIONS		
						FACING UPWARD	VERTICAL	FACING DOWNWARD
1	2	3	4	5	6	7	8	9
Fishenden and Saunders	(13)	Heated Plate	2 ft Square (Max.)	~	To Over 1000 F	$h_c^* = 0.27\ (\Delta t/L)^{0.25}$	Streamline Region $h_c^* = 0.28\ (\Delta t/L)^{0.25}$ Turbulent Region $h_c = 0.30\ (\Delta t)^{0.25}$	$h_c^* = 0.12\ (\Delta t/L)^{0.25}$
Fishenden and Saunders (McAdams)	(11)	Do.	Do.	~	Do.	$h_c^* = 0.27\ (\Delta t/L)^{0.25}$ $h_c = 0.22\ (\Delta t)^{0.33}$	Streamline Region ~ Turbulent Region ~	$h_c^* = 0.12\ (\Delta t/L)^{0.25}$ ~
Griffiths and Davis	(14)	Heated Plate	3 to 4 ft sq	Room Temperature	Up to 212 F	$h_c = 0.4\ (\Delta t)^{0.25}$	$h_c = 0.3\ (\Delta t)^{0.25}$	$h_c = 0.2\ (\Delta t)^{0.25}$
Griffiths and Davis (McAdams)	(20)	Do.	Do.	Do.	Do.	$h_c = 0.38\ (\Delta t)^{0.25}$	$h_c = 0.27\ (\Delta t)^{0.25}$	$h_c = 0.2\ (\Delta t)^{0.25}$
Heilman	(15)	Heated Disk ..... Heated Ribbon	3.47 to 9.97 in. ..... 0.50 in. Wide, to 2-108 in. Long	Air at Room Temperature	Zero to 700 F	$C = 1.79$	$h_c^{**} = C\Delta t^{0.266}/D^{0.2}T_{avg}^{0.181}$ $C = 1.394$	$C = 0.89$
Heilman (Carroll)	(19)	Do.	Do.	Do.	Do.	$h_c^{**} = 0.32\ (\Delta t)^{0.27}$	$h_c = 0.25\ (\Delta t)^{0.27}$	$h_c^{**} = 0.16\ (\Delta t)^{0.27}$
King	(16)	Correlated Data From Other Investigators				$h_c^{**} = 0.28\ (\Delta t)^{0.33}$	$h_c = 0.22\ (\Delta t)^{0.33}$	$h_c^{**} = 0.15\ (\Delta t)^{0.33}$
Nusselt and Henky (Giesecke)	(18)	~	~	~	~	$h_c = 0.48\ (\Delta t)^{0.25}$	$h_c = 0.38\ (\Delta t)^{0.25}$	$h_c = 0.22\ (\Delta t)^{0.25}$
Wilkes and Peterson	(17)	Heated Plate	8 ft by 2 ft 8 in. With 4 in. Air Space Between Surfaces	Air at About Room Temperature	8 to 28 F	$h_c = 0.83\ (\Delta t)^{0.12}$	$h_c = 0.5\ (\Delta t)^{0.12}$	$h_c = 0.38$
ASHAE Research Lab.		Environment Lab.	24½ x 12 x 12 ft. (Max.) (See Paper)	Room Temperatures	Zero to 100 F	FLOOR HEATED (From Floor or to Ceiling) $h_c = 0.39\ (\Delta t)^{0.31}/D_e^{0.08}$	TO WALLS (Floor Heated or Ceiling Heated) $h_c = 0.29\ (\Delta t)^{0.22}/H^{0.05}$	CEILING HEATED (From Ceiling or to Floor) $h_c = 0.041\ (\Delta t)^{0.25}/D_e^{0.24}$

* In Fishenden and Saunders' equations, L is the side length in feet; a mean value is taken for the side length if the plate is rectangular.

** Heilman's (References 15, 19) and King's (Reference 16) values for horizontal heated plates facing upward and downward were based on the percentage-wise distribution obtained by Griffiths and Davis (Reference 14). In Heilman's Equation, D is the diameter or length in inches and T_{avg} is the average of surface and air temperature, Fahrenheit degrees absolute. Heilman suggested that, until further experimental work was done, 24 in. be taken as the limit for the size and height, and he assumed the heat loss becomes constant for heights greater than 24 in.

Figs 2a and 2c), the characteristic linear dimension was concluded to be the equivalent diameter for the heated floor and ceiling panels. A similar correlation for the convection to the walls, as shown in Fig. 2b, indicated that the characteristic linear dimension of a wall is its height.

The importance of the characteristic dimension in Equation 4 depends upon the exponent of the equation. If the exponent is 1/3 ,then the characteristic dimension appears on both sides of the equation as $L = (L^3)^{1/3}$ and consequently L cancels, indicating that the convection coefficient is independent of the size of the surface. It will be noted that in floor panel heating (Equation 5), the exponent is nearly 1/3 and the effect of the dimension D_e is very small (see Equation 6). The close agreement of the convection values obtained by other investigators for various sizes of plates as shown in Fig. 4, may be attributed to this fact. In contrast, however, the characteristic dimension is relatively important in determining convection heat transfer from a heated ceiling panel, as shown by Equation 12, since the exponent for this orientation is ¼ (see Equation 11).

In Equation 7 for wall convection with a heated floor panel, the exponent of 0.32 indicates that the characteristic dimension (wall height) is not important. There is some evidence (Equation 13 and Fig. 4) that the coefficient is considerably different for wall convection with a heated ceiling, and that the wall height in this case may be an important factor. However, over the range of conditions covered in this paper it is suggested that Equation 8 be used for wall convection with either floor or ceiling heating.

CONCLUSIONS

The data reported in this paper were obtained with the entire floor area or ceiling area used as a heated panel, a uniform environment wherein all surfaces other than the heated panel were at a uniform temperature, relatively still air conditions (no infiltration), and an empty, unlighted room. Under these conditions the following conclusion may be drawn:

1. On the basis of research done at the ASHAE Laboratory, the following equations apply in calculating natural convection heat transfer at room surfaces.

A. In a floor-heated space

1. Convection from floor: $q_c = 0.39\ (\Delta t)^{1.31}/D_e^{0.08}$
2. Convection to walls: $q_c = 0.29\ (\Delta t)^{1.32}/H^{0.05}$
3. Convection to ceiling: Same as convection from floor.

B. In a ceiling-heated space

1. Convection from ceiling: $q_c = 0.041\ (\Delta t)^{1.25}/D_e^{0.24}$
2. Convection to walls: Same as for floor-heated space.
3. Convection to floor: Same as convection from ceiling.

2. Natural convection data given by other investigators for small heated plane surfaces were found to be in good agreement with all of the equations listed in the first conclusion except the equation for natural convection from a heated ceiling. The convection coefficient for small free-edge plates may be 6 to 10 times as great as that for a heated ceiling.

3. Room size in a ceiling-panel-heated room has a significant effect on the unit convection from the ceiling and the floor. However, the convection heat transfer from a heated ceiling is small in comparison with radiation exchange, and therefore, the effect

of room size on the total heat transfer is not important. In a floor-heated room, the effect of room size is not significant.

4. In a completely enclosed space with high emissivity room surfaces, the interchange factor between the heated panel and its enclosure may be approximated by the hemispherical emissivity of the panel surface.

5. The approximate combined film conductances (natural convection and radiation) for the heated panels based on the difference between panel temperature and the room air temperature for a normal size room with high emissivity surfaces are as follows:

$h_{ro} = 2$ for heated floor panel at about 85 F.

$h_{ro} = 1.1$ for heated ceiling panel at about 120 F.

ACKNOWLEDGMENT

The authors acknowledge the suggestions and assistance of members of the staff of the ASHAE Research Laboratory and especially the members of the TAC on Panel Heating and Cooling.

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APPENDIX

THERMAL REFLECTING NATURE OF A PAINTED SURFACE

Radiant heat transfer between surfaces is influenced by the nature of the reflection of radiation from the surfaces in question and their surroundings. A surface may reflect a radiant thermal beam perfectly diffusely*, or specularly**, or in any pattern between these 2 limits. In all cases, the reflectivity will vary with the angle of the incident beam. The painted surfaces of the floor and heat-flow meters located on the floor panel were found to behave as a highly specular rather than a perfectly diffuse reflector by the use of a spot heat source. The highly specularly reflecting characteristics of the painted surface were also confirmed by comparing the radiometer measurement data with the calculated values for specular and perfectly diffuse reflectors as shown in Fig. A-1 for a uniform environment. The method of employing a spot heat source, the radiometer measurements, and the computation of the net radiation exchange between a perfectly diffuse reflector and surrounding, and between a specular reflector and surrounding are described in the following paragraphs.

In the use of the spot heat source, a radiometer[6] was sighted at a given inclination at an area on the painted floor. The spot heat source was sighted at the same area, but was located at a 90 deg horizontal angle to the radiometer so that only diffusely reflected radiation could enter the radiometer. The heat source was then positioned opposite the radiometer so that both specularly and diffusely reflected energy would pass into the instrument. In either case, the radiometer was unable to *see* the heat source directly. Higher radiometer outputs were observed in the latter than in the

* A diffusely reflecting surface converts the incident beam into a bundle of beams distributed into all directions over the hemisphere above the surface. A perfectly diffusely reflecting surface distributes the reflected beams not only in all directions but also according to the cosine law.

** A specular reflecting surface reflects the impinging beam as a defined and limited beam at an angle to the angle of incidence.

former arrangement. This indicated that the thermal reflecting nature of the painted surface was highly specular.

The experimental data shown in Fig. A-1 were obtained with the radiometer by measuring the net radiation exchange between the painted surface of the heated floor and the walls and ceiling at various angles. The radiometer was first sighted at the painted surface of the floor at a given angle and measured the radiation exchange between its thermopile and the heated surface. It was then turned to the opposite direction and measured the radiation exchange between the thermopile and the surroundings. The difference of the 2 observed values, i.e., the energy leaving the heated surface and the

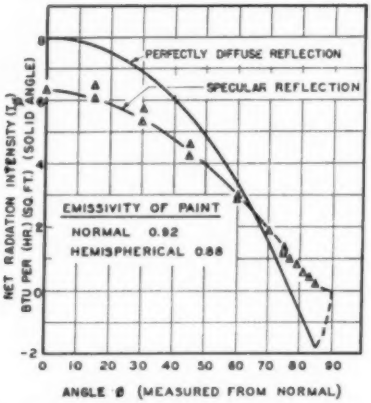

FIG. A-1—NET RADIATION INTENSITY FOR A HEATED PANEL VS. ANGLE OF INCIDENCE (FLOOR 85 F, AUST 65 F, UNIFORM ENVIRONMENT, NO INFILTRATION, 24½ X 12 X 8 FOOT ROOM)

energy impinging on the surface, is the net radiation exchange between that surface and the surroundings at that angle.

The computation of the net radiation intensity between the painted surface of the heated floor and the walls and ceiling, at various angles and in a uniform environment, was based on Equations A-1 to A-4. The painted surface was first assumed as a perfectly diffuse reflector and then as a specular reflector.

For a perfectly diffuse reflector,

$$I_{\phi net} = e_\phi \sigma \frac{T_s^4}{\pi} \cos \phi + r_h \sigma \frac{T^4_{AUST}}{\pi} \cos \phi - \sigma \frac{T^4_{AUST}}{\pi} \cos \phi \quad . \quad . \quad . \quad . \quad . \quad (A\text{-}1)$$

or

$$I_{\phi net} = \frac{\sigma}{\pi} [e_\phi T_s^4 - e_h T^4_{AUST}] \cos \phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (A\text{-}2)$$

For a specular reflector,

$$I_{\phi net} = e_\phi \sigma \frac{T_s^4}{\pi} \cos \phi + r_\phi \sigma \frac{T^4_{AUST}}{\pi} \cos \phi - \sigma \frac{T^4_{AUST}}{\pi} \cos \phi \quad . \quad . \quad . \quad . \quad . \quad (A\text{-}3)$$

or

$$I_{\phi net} = e_\phi \frac{\sigma}{\pi} [T_s^4 - T^4_{AUST}] \cos \phi \quad \ldots \ldots \ldots \ldots \ldots \quad (A\text{-}4)$$

where $I_{\phi net}$ = Net intensity at an angle ϕ from the normal to the surface, Btu/(hr) (sq ft) (solid angle)
e_ϕ = Angular emissivity[6] of the surface at an angle ϕ from the normal, dimensionless
e_h = Hemispherical emissivity[6] of the surface, dimensionless.
$r_\phi = 1 - e_\phi$ = Angular reflectivity of the surface at an angle ϕ from the normal, dimensionless
$r_h = 1 - e_h$ = Hemispherical reflectivity of the surface, dimensionless
T_s = Absolute temperature of the surface, deg R
T_{AUST} = Absolute temperature of the AUST, deg R
ϕ = Angle made by a thermal radiant beam with normal to surface, dimensionless
σ = Stefan-Boltzmann constant, 0.173 x 10^{-8} Btu per (hr)(sq ft) (deg R^4), in solid angle 2π, above a surface
π = Solid angle expressed in steradians, dimensionless

The assumptions by which the equations were established are:

1. The total energy leaving a surface per unit time per unit area in 2π solid angle above a surface, usually defined as radiosity[10] is equal to the sum of the energies leaving by emission, reflection, and transmission. The net energy exchange between the heated floor and the surroundings is the difference of energy leaving the floor and surrounding.

2. The surface is opaque or non-transmitting, i.e., the transmissivity equals zero. Kirchhoff's law is valid: the absorptivity of a surface is numerically equal to its emissivity. The reflectivity, r, is then dependent only upon the emissivity, e, i.e., $r = 1 - e$.

3. The normal intensity I_n of emission of a black body is the quantity of energy emitted by a perfectly diffuse surface per unit area through unit solid angle in a direction normal to the surface, and may be expressed as

$$I_n = \frac{\sigma T_s^4}{\pi}$$

The intensity of emission by a perfectly diffuse surface at an angle ϕ to the surface normal, I_ϕ, may be represented by

$$I_\phi = I_n e_\phi \cos \phi$$

4. The intensity of irradiation from all directions to the floor panel is equivalent to the black body irradiation at the temperature of the walls and ceiling, i.e.,

$$\sigma \frac{T^4_{AUST}}{\pi}.$$

When the heated floor panel was at 85 F and the walls and ceiling at 65 F, radiometer measurements showed that the intensity of irradiation to the floor was the same from all directions for the angles measured, and that the quantity of irradiation was essentially the same as the black body emission at 65 F.

5. The intensity of the reflected radiation from the floor, at an angle ϕ to the surface normal, due to the irradiation from the walls and ceiling, is $r\sigma \frac{T^4_{AUST}}{\pi} \cos \phi$. If it is a perfectly diffuse reflector, this becomes $r_h\sigma \frac{T^4_{AUST}}{\pi} \cos \phi$. If it is a

specular reflector, the intensity becomes

$$r_\phi \sigma \frac{T^4_{AUST}}{\pi} \cos \phi.$$

The curves for perfectly diffuse and specular reflectors were constructed for a floor temperature of 85 F and wall and ceiling temperatures of 65 F. The values of angular and hemispherical emissivity of the painted surface were taken from reference (6). The experimental data were found to be closer to the curve for a specular reflector. The highly specularly reflecting nature of the painted surface was therefore indicated.

DISCUSSION

P. B. GORDON, New York, N. Y.: I do not propose to discuss the technical aspect of the paper. I would like to comment, though, that the information presented results from development work on an earlier phase of the program. Most of the original work on heat transfer within the room had to do with gross outputs. This paper, using the earlier data, separates the radiation and convection components and compares them with previously published information.

However, it is important to remember that the convection coefficients presented here are for a room held at a single uniform temperature for all surfaces and without influence of infiltration.

It would be extremely valuable to have this development carried forward for rooms with walls at non-uniform conditions, that is, an extremely cold wall adjacent to a heated ceiling with a fair amount of infiltration. It could be that the values would be somewhat different.

T. J. BLACK, Detroit, Mich.: I would like to ask the authors if they have experimented in high temperature hot water heating on radiant ceiling panels. Particularly I am thinking of the high pressure temperature systems with water temperatures around 300 or 350 F, and whether the same percentage of convection follows for the higher water temperatures in the ceiling panels.

MARTIN CHRISTESEN, Englewood, N. J.: I would like to learn from the authors what determined the ceiling temperature and the ceiling height in comparison with the 85F floor temperature. Was it because they wanted the same gross output to get their convection coefficient?

N. B. HUTCHEON, Ottawa, Canada: The authors have arrived at coefficients for combined radiation and convection of 2 and 1.1 for floor and ceiling respectively, giving a difference of 0.9. On the assumption that radiation is approximately the same in either case one would expect the difference between the total coefficients to be roughly the same as the difference between the convection coefficients which are given as 0.7 and 0.05, with a difference only of 0.65. I wonder if the authors would care to comment on this.

R. L. MAHER, Waterbury, Conn.: I would like to know more about the convection from a heated wall. We seem to be complete in other phases of the work and that data is needed to revise THE GUIDE. That is one part that is missing in the coverage.

C. M. HUMPHREYS, Cleveland, Ohio: The information given in this paper is of interest to members of ASHAE because of its direct application to panel heating. The paper should also be of interest to many others working in the field of heat transfer, for it contains hitherto unavailable data on the effect of surrounding walls or baffles on convection from a heated surface facing downward.

K. T. WHITBY*, Minneapolis, Minn.: There is one thought that occurred to me in connection with these very interesting results on the ceiling and that is whether the presence of a cold wall on one side of a room and a warmer wall on the other side might cause a convection pattern that might raise those convection coefficients on the ceiling somewhat.

LINN HELANDER, Manhatten, Kans.: It has occurred to me that if the data on heat radiation and convection from ceilings is in error, it may be that the heat gain through roofs may also be in error. I wonder if that might be the case.

AUTHORS' CLOSURE (T. C. Min and L. F. Schutrum): Mr. Gordon pointed out the limitation in the use of the data presented in the paper and stressed the need to have the analysis carried forward for rooms with walls at non-uniform conditions. It is always gratifying to hear such encouragement.

Mr. Black asked if experiments of high temperature hot water such as 350 F had been made in the ceiling-panel testing. The work at the Laboratory on panel heating covered a temperature range up to 150 F but no higher.

In answer to Mr. Christesen's question, tests in the Environment Laboratory covered the full range of panel and unheated surface temperatures encountered in practice. Floor panel temperatures were varied from 75 F to 110 F and ceiling panel temperatures from 90 F to 150 F. During both series of tests, the AUST was varied from 40 F to 70 F. Tests were also made with 3 different sized rooms, 24½ x 12 x 8 ft, 24½ x 12 x 12 ft, and 12 x 12 x 8 ft. There was no attempt to compare them at all.

Dr. Hutcheon pointed out that the difference of combined coefficients for a heated floor and a heated ceiling is 0.9, yet the difference in convection coefficients between the 2 above-mentioned cases is only 0.65. The combined film conductance (i.e. combined convection and radiation coefficient) for a surface is based on the temperature difference between the surface and the room air and is obtained by dividing the total heat flow from the surface to the room air and its surrounding by this temperature difference. The laws governing thermal radiant and convective transfer are different. The potential or driving force in convection is related to the temperature difference of a surface and the contacting fluid, while in radiation it is the difference of the fourth power of the absolute temperatures of a surface and its surrounding. Moreover, the temperature differences between the floor surface and room air and other surfaces in a floor-heated room are much lower than the temperature differences between the heated ceiling and the air and other surfaces in a ceiling-heated room. Therefore, it is hardly possible to make a direct comparison of the combined film conductances between the 2 cases. They are actually fictitious coefficients only serving for convenience in design calculation, and should be broken down into a radiation coefficient, and to that the convection coefficient should be added.

Mr. Maher was interested in knowing about the specific data in a wall-panel heated room. It has been recognized that the work in panel heating at the Laboratory is not complete without including the information on wall heating. At present, only very limited information is available. For instance, in a ceiling cooled space with one warm wall, the convection from the wall could be approximated by using the convection equation for heat flow to the walls in a floor-heated space.

In Dr. Whitby's discussion, it was mentioned whether the convection from the heated ceiling would increase in the presence of a cold wall on one side and a warmer wall on the other side of a room. Recently, the Laboratory undertook a study of cold-wall effect on a ceiling-panel-heated room, in which the conditions are very close to what Dr. Whitby described—a heated ceiling, a cold wall, and warm walls. There was actually considerable air flowing down the cold wall, traveling across the floor, and flowing up the opposite warm wall. Under these conditions, the convection coefficients

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* University of Minnesota.

for the ceiling increased. The total output from the ceiling, however, did not vary appreciably.

Professor Helander commented about the error in heat gain through roofs since the data previously used for convection from a heated ceiling may be in error. In the recent GUIDE, the total heat transfer from the ceiling was revised to the extent that both heat flow up (winter condition) and heat flow down (summer condition) are included. In this paper, however, only the heat transfer from the room-side surface to the room was considered rather than what was behind the surface.

1577

No. 1577

HEAT REQUIREMENTS OF SNOW MELTING SYSTEMS

By W. P. CHAPMAN* AND SAMUEL KATUNICH†, PITTSBURGH, PA.

A SNOW melting research project was initiated in 1952 at the National Tube Division, U. S. Steel, Research Laboratory for a dual purpose. First, to substantiate experimentally a proposed equation for operating heat requirements, and second, to determine the heat requirement of a system during the *idling* period prior to snowfall.

The proposed equation is based on the *free-area* concept[1], and includes the insulating effect of the un-melted snow. The *free area ratio* has been defined[1] as the ratio of the area *free* of snow to the total area. It may be assumed that any area even partially covered by snow would have an insulating effect provided by the unmelted snow. It may be said then, that the *free area ratio*, A_r, represents an insulating factor.

The proposed equation has been substantiated by experimental data and can be written as:

$$q_t = q_s + q_m + A_r (q_e + q_h) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

with symbols as defined in Nomenclature and with its derivation demonstrated in Appendix B. In Appendix B also it is shown that the value of A_r can be expressed as

$$A_r = [q_t - (q_s + q_m)]/[q_0 - (q_s + q_m)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The 4 heat requirements: (1) sensible heat gain of snow, q_s, (2) latent heat of fusion, q_m, (3) latent heat of evaporation, q_e, and (4) heat transfer, q_h, are equated in Appendix A and have been derived in the literature[1].

It may be shown that there are just 3 practical values for A_r, namely, zero, 0.5 and 1.0. When the free area ratio, A_r, is equal to zero, the slab is covered completely with snow. Although this condition can be defined as a failure of the system, it may be tolerable in some instances for short periods of time. Examples would be residential driveways or walks where time for clearance is not critical.

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† Metallurgical Research Engineer, National Tube Division, United States Steel Corp.
[1] Design of Snow Melting Systems, by W. P. Chapman (*Heating and Ventilating*, April, 1952).
Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956.

When the free area ratio, A_r, is equal to unity, the slab is completely free of snow. This condition requires the maximum energy supply to the slab. Such a design is warranted for emergency areas or any area where snow accumulation cannot be tolerated.

An intermediate value of $A_r = 0.5$ can be used for less critical areas, such as sidewalks around commercial buildings. It will be shown that a very light cover-

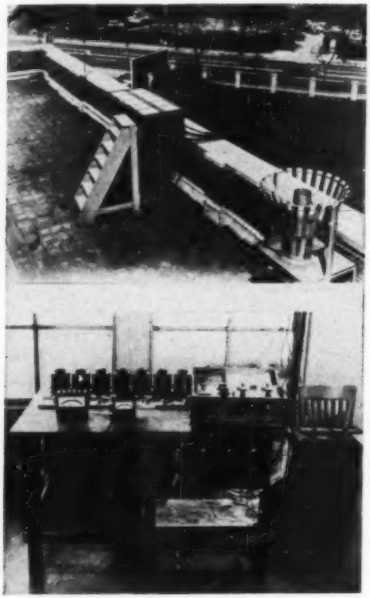

FIG. 1—(TOP) THE LOCATION OF THE PANELS AND WEATHER EQUIPMENT. (BOTTOM) THE POWER INPUT AND THERMOCOUPLE CONTROLS

age of snow will develop a free area ratio of 0.5, but probably will not reduce traction significantly.

The second part of the research project dealt with the *idling* requirements of a snow melting system. Information was sought to answer the question of the heat required to maintain a slab surface temperature above freezing during periods of no snowfall. It was found that the convection losses in Btu per hr (sq ft) can be estimated by the equation:

$$q_{bc} = (0.27\ v + 2.5)(t_p - t_a). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Since most systems are idled during overcast periods with air temperatures averaging between 20 and 25 F, it is felt that sufficient accuracy is obtained by using a

radiation coefficient of 0.8 in Btu per (hr) (sq ft), which gives an *idling* equation of

$$q_b = (0.27\ v + 3.3)(t_p - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

This equation represents the convection and radiation transfer from a dry slab to an ambient temperature equal to the air temperature.

DESCRIPTION OF EQUIPMENT

Two different types of snow melting panels were used. The first type, shown in Fig. 1, consisted of 10 one-foot square panels made up of insulated nichrome heat-

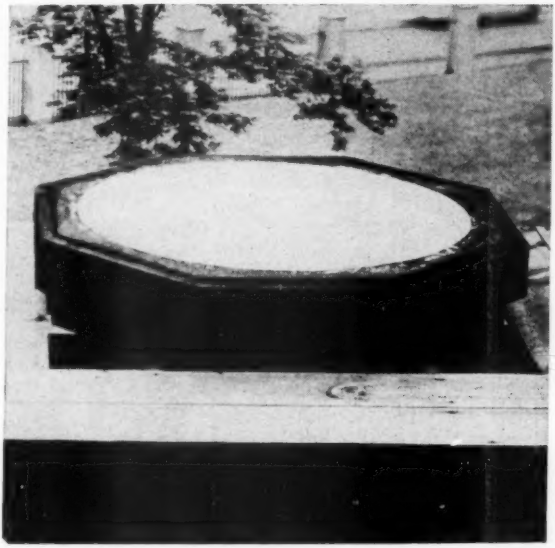

FIG. 2—A VIEW OF THE ROUND PANEL OF 10 SQUARE FEET AREA—LOCATION APPROXIMATELY 15 FT LEFT OF THE PANELS SHOWN IN FIG. 1

ing elements spaced on ¾ in. centers under ½ in. of cement mortar. The 10 panels were insulated from each other by 2 in. wood separators. Two-inch thick foam glass bricks at the sides and a 6-in. layer of fiber glass wool under the panels prevented excessive heat losses in those directions. A slope of ¼ in. per ft was provided to allow for water run-off. To protect the apparatus from water penetration, all electrical outlets and all insulating materials were waterproofed.

At least 2 copper-constantan thermocouples per panel were installed to measure panel surface temperatures. The thermocouple beads were thin copper rectangles 0.03 x 0.40 x 0.25 in. placed on a center line parallel and as close to the panel surface as possible without exposing them to the atmosphere. It was thought that

having a plate as the hot junction would eliminate the effect of localized chilling. Ordinary thermocouples were used to measure insulation temperatures.

The second type of panel, shown in Fig. 2, was a round panel having an area of 10 sq ft. Its heating elements and insulation were similar to those of the square panels. This round panel was used primarily for the *idling* tests because of its uniform wind path length regardless of wind direction.

The power input to the panels was adjusted by the use of variable transformers. The power used was measured either by an ammeter-voltmeter setup or with a

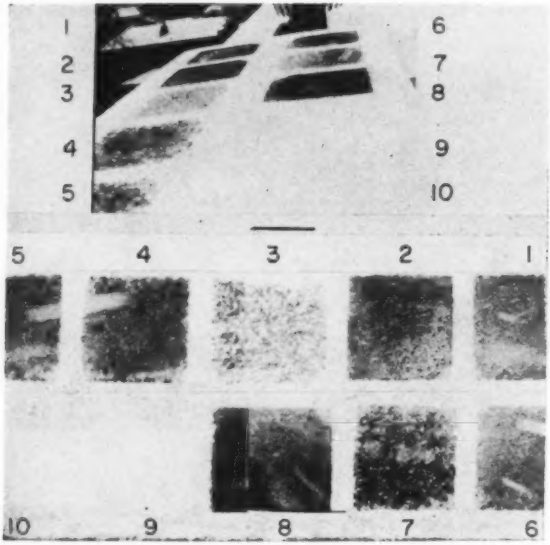

FIG. 3—PHOTOGRAPHS OF THE PANELS UNDER EQUILIBRIUM CONDITIONS WHEN INSULATION EFFECT OF SNOW WAS BEING CONSIDERED. (TOP) SIDE VIEW. (BOTTOM) TOP VIEW

wattmeter. Provisions were made through a switchboard and potentiometer arrangement to read the 45 thermocouples used.

Weather instruments used consisted of a totalizing anemometer, a recording snow gage, and a sling psychrometer. The snow gage was supplemented by weighing snowfall collected in wide-mouthed containers. This was done when the snowfall was accompanied by gusty, high winds or when relatively little snowfall occurred. The dry bulb of the psychrometer was kept free of snowflakes by enclosing it within a perforated metal sleeve.

TEST PROCEDURE

In running the tests to determine the effect of snow insulation, the power input to the panels was varied so that different thicknesses of snow were maintained upon

the panels under equilibrium conditions. Wind velocity readings and photographs of the panels were taken at the beginning and end of each chosen interval. Snowfall was measured by a recording snow gage and also by weighing the amount collected in wide-mouthed containers. Relative humidity of the atmosphere was ascertained by the use of a sling psychrometer at frequent intervals during the test.

On days when no snowfall occurred, a study of the system's idling requirements was made. Under equilibrium conditions, such as would very nearly exist after the system was idling a short time, convection and radiation losses account for all

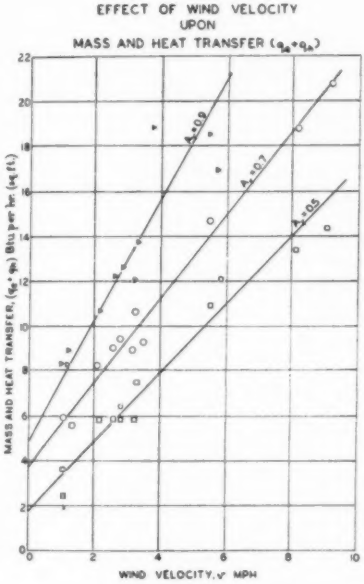

FIG. 4 (*left*)—EFFECT OF WIND VELOCITY UPON MASS AND HEAT TRANSFER ($q_e + q_h$)

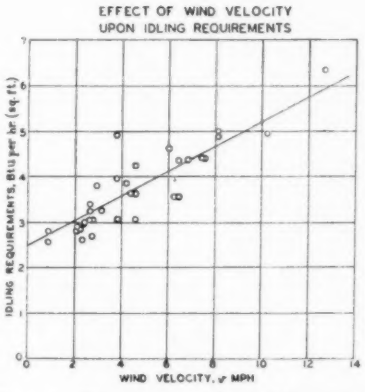

FIG. 5—EFFECT OF WIND VELOCITY UPON IDLING REQUIREMENTS

heat input except small heat losses through the insulating materials. Because of the small temperature differences involved in the idling tests, these insulation losses were considered negligible.

To account for the radiation exchange with the atmosphere, the idling tests were made under conditions where the unheated panel surface temperature was equal to or less than the air temperature. When t_p was less than t_a, one of the 10 panels was supplied with enough energy to raise its temperature to air temperature. This quantity of energy was then subtracted from the total energy supplied to the other panels. This energy difference represents the convection between the heated panels and the air, and is defined as the convection requirement.

RESULTS

Of the four heat requirements, vaporization (q_e), convection and radiation (q_h), sensible heat of snow (q_s), and the heat of fusion of snow (q_m), a saving can be re-

alized only upon the heat requirements for vaporization, convection, and radiation. Regardless of any snow insulation effect, all the snow must be raised to a temperature of about 32 F and then melted. Stated algebraically, this is Equation 1, *viz*:—

$$q_t = q_s + q_m + A_r\,(q_c + q_h)$$

Analysis of the experimental data for one test, as shown by the sample calculation in Appendix C, gives $A_r = 0.54$. A photograph of how a panel looks with a free area ratio of 0.54 is given in panel 5 of Fig. 3.

Examination of photographs of each test indicates that for all practical purposes, the only intermediate value of A_r that is significant is $A_r = 0.5$.

Results of 34 tests are plotted in Fig. 4 and lines of constant values of A_r have been fitted to the points by the method of least squares.

The other phase of the project concerned the *ilding* or *warming-up* heat requirements prior to a predicted snowfall. During this period, all heat transfer from the slab surface is by radiation or convection. It was initially assumed that idling requirements could be expressed in an exponential equation which would include as variables both the wind velocity and the difference in air and panel temperature. However, the tests made indicate that within the range of atmospheric conditions during each test run, the wind velocity and air temperature can be included in a *linear* equation to represent the idling requirements.

This linear equation (in Btu per hour per square foot) is of the form:

$$q_b = (f_c + f_r)(t_p - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The coefficient for radiation, f_r, can be taken as

$$f_r = 0.173\,\frac{\left(\frac{T_p}{100}\right)^4 - \left(\frac{T_a}{100}\right)^4}{t_p - t_a} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)\dagger$$

Several solutions of Equation 6 are given in Table 1.

From Table 1 it can be seen that $f_r = 0.8$ is a sufficiently accurate assumption for usual air temperatures.

Fig. 5 is a plot of the experimental data for the convection coefficient, f_c. The slope and intercept of the line of best fit (fitted by the method of least squares) in Fig. 5 gives the coefficient a value of

$$f_c = 0.27\,\dot{v} + 2.5$$

The values of f_r and f_c can then be substituted in Equation 5 as follows:

$$q_b = (f_c + f_r)(t_p - t_a)$$

$$q_b = (0.27v + 2.5 + 0.8)(t_p - t_a)$$

$$q_b = (0.27v + 3.3)(t_p - t_a)$$

and becomes Equation 4, evaluated for practical use, as the output for a dry, bare slab, or the heat requirement for an idling period.

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† In Equation 6, T_a and t_a are used as if they were the *equivalent receiver temperature* which is the weighted average temperature including various surface temperatures and *sky* temperature, such that if the snow-melting panel were enclosed by an isothermal hood, with shape and emissivity factors of unity, the radiation exchange would be the same as that found in the field. This use of T_a and t_a is within the accuracy of the paper.

TABLE FOR PRACTICAL USE

In order to simplify the application of the data, Table 2 has been prepared. It may be used directly to determine the heat outputs and fluid temperatures needed for the usual ranges of air temperature, wind velocity, relative humidity, and snowfall rates for slabs designed to be completely free of snow, ($A_r = 1$) and completely covered, ($A_r = 0$). It may also be used to determine the heat required for the intermediate case when $A_r = 0.5$. This use is illustrated by the following procedure.

NOMENCLATURE

A_r = free area ratio.
f_c = film coefficient of heat transfer for convection, Btu per (hour) (square foot) (Fahrenheit degree).
f_r = film coefficient of heat transfer for radiation, Btu per (hour) (square foot) (Fahrenheit degree).
h_{fg} = enthalpy of saturated vapor at film temperature, Btu per pound.
P_{av} = vapor pressure of moist air, inches of mercury.
P_{wv} = vapor pressure of water, inches of mercury.
q_b = heat transfer from a bare panel (idling conditions) Btu per (hour) (square foot).
q_{bc} = heat transfer by convection from a bare panel, Btu per (hour) (square foot).
q_e = heat of vaporization, Btu per (hour) (square foot).
q_h = heat transferred by convection and radiation, Btu per (hour) (square foot).
q_i = total heat input, Btu per (hour) (square foot).
q_l = heat loss through insulation (edge and bottom losses in terms of panel area), Btu per (hour) (square foot).
q_m = heat of fusion, Btu per (hour) (square foot).
$q_0 = q_e + q_m + q_s + q_{hr}$ Btu per (hour) (square foot).
q_s = sensible heat gain of snow, Btu per (hour) (square foot).
q_t = total heat output, Btu per (hour) (square foot).
s = rate of snowfall, inches of water equivalent per hour (the unit, inches of water equivalent per hour, is defined as the depth of water that would be obtained if the snowfall per hour were melted).
t = temperature of the circulated fluid for anti-freeze solution.
t_a = air temperature, Fahrenheit.
T_a = air temperature, Fahrenheit absolute.
t_p = panel surface temperature, Fahrenheit.
T_p = panel surface temperature, Fahrenheit absolute.
t_f = water film temperature, Fahrenheit.
v = wind velocity, miles per hour.

Problem: Find the heat requirement of an installation when the snowfall rate is 0.16 ($s = 0.16$); *the air temperature,* t_a, *is* 10 *F; the wind velocity,* v *is* 10 *mph, the relative humidity is* 75 *percent; and where a partially free surface can be tolerated* ($A_r = 0.5$).

Solution: From Table 2 note that for the stated conditions the value of q_t for $A_r = 1.0$ is 234 Btu per (hr) (sq ft). Also that for $A_r = 0$, the value of q_t is 129 Btu per (hr) (sq ft). For the intermediate value of $A_r = 0.5$, the requirement is

$$0.5\ (234 + 129) = 181 \text{ Btu per (hr) (sq ft).}$$

TABLE 1—VALUES OF RADIATION COEFFICIENT, f_r, AT USUAL AIR TEMPERATURES

AIR TEMPERATURE, t_a	RADIATION COEFFICIENT[a], f_r, BTU PER HR (SQ FT) (F DEG)
30	0.819
25	0.807
20	0.795
15	0.782

[a] For a panel surface temperature of t_p = 32 F.

TABLE 2—HEAT OUTPUT AND FLUID TEMPERATURES FOR SNOW MELTING SYSTEMS

RELATIVE HUMIDITY, PERCENT	RATE OF SNOWFALL, INCHES OF WATER EQUIVALENT PER HOUR	FREE AREA RATIO[a] A_f		t_a AIR TEMPERATURE, FAHRENHEIT											
				0			10			20			30		
				v, WIND VELOCITY, MPH											
				5	10	15	5	10	15	5	10	15	5	10	15
75	0.08	1.0	q_t[b]	151	206	260	128	169	210	103	129	156	76	87	97
			t	108	135	163	97	118	138	85	98	111	71	76	81
		0.0	q_t	66	66	66	64	64	64	62	62	62	60	60	60
			t	66	66	66	65	65	65	64	64	64	63	63	63
	0.16	1.0	q_t	218	273	327	193	234	275	166	192	219	136	147	157
			t	142	169	197	129	150	170	117	130	143	101	106	111
		0.0	q_t	133	133	133	129	129	129	125	125	125	121	121	121
			t	99	99	99	97	97	97	95	95	95	93	93	93
	0.25	1.0	q_t	293	347	402	265	306	347	236	262	289	204	214	224
			t	179	207	234	166	186	207	151	164	177	135	140	145
		0.0	q_t	208	208	208	202	202	202	195	195	195	188	188	188
			t	137	137	137	134	134	134	131	131	131	127	127	127
90	0.08	1.0	q_t	150	204	258	126	167	208	100	125	150	72	80	87
			t	108	135	162	96	116	136	84	96	109	69	72	76
		0.0	q_t	66	66	66	64	64	64	62	62	62	60	60	60
			t	66	66	66	65	65	65	64	64	64	63	63	63
	0.16	1.0	q_t	217	271	325	191	232	273	163	188	213	132	140	147
			t	142	169	198	128	148	168	116	128	141	99	102	106
		0.0	q_t	133	133	133	129	129	129	125	125	125	121	121	121
			t	99	99	99	97	97	97	95	95	95	93	93	93
	0.25	1.0	q_t	291	346	400	264	304	344	233	258	283	200	208	215
			t	179	206	233	165	185	205	150	162	175	133	137	141
		0.0	q_t	208	208	208	202	202	202	195	195	195	188	188	188
			t	137	137	137	134	134	134	131	131	131	127	127	127

[a] Free Area Ratio, A_f = snow-free area ÷ total area.
[b] q_t = heat output, Btu per (hour) (square foot); t = fluid temperature, Fahrenheit (based on ¾-in. pipe on 12-in. centers with 2-in. concrete cover.)

CONCLUSIONS

Data obtained as a result of this investigation and the observations made while performing the snow melting operation indicate the following conclusions:

1. If a slight amount of snow is allowed to accumulate upon a snow melting panel, there is an insulation effect dependent upon the existing wind velocity, relative humidity, and air temperature. Conditions of a high wind velocity and a low air temperature offer the greatest heat savings potential. Relative humidity is the least important.

2. An A_r of 0.5 results in considerable heat saving and involves a snow covering that could be tolerated in many non-critical installations. An A_r of less than 0.5, possibly one of zero, would not be justified unless snow accumulation were acceptable for short periods of time.

3. To calculate approximate heat requirements to pre-warm a system, Equation 4 can be used if the wind velocity is not greater than 12 mph.

4. The basic Equation for a snow melting system can be taken as Equation 1.

5. Table 2 can be used to determine the basic design requirements for snow melting systems over the usual design range.

APPENDIX A

BASIC EQUATIONS FOR CALCULATING HEAT REQUIREMENTS

The following 5 equations are those used in calculating the various heat quantities which are needed in arriving at the total heat requirement.

Heat of Vaporization, q_e: This quantity can be found from the expression:

$$q_e = (av + b)\,(P_{wv} - P_{av})h_{fg} \quad \text{(A-1)}$$

where

$a = 0.0201$
$b = 0.055$

Sensible Heat Gain of Snow, q_s: Use of the following expression yields the sensible heat quantity:

$$q_s = 2.6\,(t_f - t_a)s \quad \text{(A-2)}$$

Heat of Fusion of Snow, q_m: can be calculated from the expression

$$q_m = 746s \quad \text{(A-3)}$$

Heat Transferred by Convection and Radiation, q_h: This quantity is determined by applying the expression:

$$q_h = 11.4\,(av + b)\,(t_f - t_a) \quad \text{(A-4)}$$

where

$a = 0.0201$
$b = 0.055$

Equations A-1, A-2, A-3, and A-4 are derived in Reference 1, and all are expressed in the units stated in the Nomenclature.

Total Heat Requirements, q_t: This is the quantity found by Equation 1 of this paper, *viz:*

$$q_t = q_s + q_m + A_r(q_e + q_h) \quad (1)$$

and which is derived in Appendix B.

APPENDIX B

DERIVATION OF EQUATIONS 1 AND 2

Definitions: It is necessary first to set up definitions of the terms used. First, let

A_f = area *free* of snow, in square feet so that the total output from the *free* part of the panel would be: $q_0 = q_e + q_h + q_m + q_s$

Then as a second definition, let

A_c = area *covered* by snow, in square feet so that the total output from the *covered* part of the panel would be $q_s + q_m$. Also let $A_t = A_f + A_c$, and further let $A_r = A_f/A_t$ = *free area* ratio.

Since

$$A_t = A_f + A_c,$$

then

$$1 = (A_f/A_t) + (A_c/A_t) = A_r + (A_c/A_t)$$

and

$$A_c/A_t = 1 - A_r$$

Derivation: From test data

$q_t = q_1 - q_i$ = total heat *output* per unit area.

Hence,

$A_t q_t$ = total heat output from entire panel.

But from the definitions

$A_f q_0 + A_c(q_s + q_m)$ = total heat output from the panel.

Therefore

$$A_t q_t = A_f q_0 + A_c(q_s + q_m)$$

Dividing by A_t

$$q_t = A_r q_0 + (1 - A_r)(q_s + q_m)$$

and

$A_r = [q_t - (q_s + q_m)]/[q_0 - (q_s + q_m)]$, which is Equation 2 in the text.

Limits: There are three limiting cases which need to be considered, *viz.*:

Case I, when $A_r = 1$, and surface is wholly free of snow.
Then

$$q_t = q_0 = q_e + q_h + q_m + q_s$$

and

$$A_f = A_t$$

This can be checked by noting that:

$$1 = [q_0 - (q_s + q_m)]/[q_0 - (q_s + q_m)] = 1$$

Case II, when $A_r = 0$, and surface is wholly covered by snow.
Then

$$q_t = q_s + q_m$$

and

$$A_e = A_t$$

This case can be checked by noting that:

$$0 = [q_s + q_m - (q_s + q_m)]/[q_0 - (q_s + q_m)]$$

Case III, when A_r lies between the values of 1.0 and zero, and the surface is partially covered by snow.
Then

$$q_t = q_s + q_m + A_r(q_e + q_h)$$

In this case the value of A_r is:

$$A_r \frac{[q_t - (q_s + q_m)]/[q_0 - (q_s + q_m)]}{[q_0 - (q_s + q_m)]} = q_t - (q_s + q_m)$$

but

$$q_0 = q_e + q_h + q_s + q_m, \text{ so } q_0 - (q_s + q_m) = q_e + q_h$$

Then

$$q_t = q_s + q_m + A_r\,(q_e + q_h),$$

which checks and is Equation 1 in the text.

APPENDIX C

SAMPLE CALCULATIONS FOR APPLYING BASIC EQUATIONS OF APPENDIX A TO EXPERIMENTAL DATA TO DETERMINE A_r

The calculations are based upon the assumption that equilibrium conditions exist, that the snow is at air temperature, and that the snow forms a 0.01 ft film of water when it melts.

Problem: Determine the value of A_r for the case of the panel 5 as illustrated in Fig. 3, where the snow is partially melted, and which produced the following *experimental*: data: $t_p = 34.2$; $t_a = 29.2$; $s = 0.048$; $v = 2.1$ mph; relative humidity = 88 percent (so $P_{av} = 0.142$); $q_i = 43.7$, and $q_t = 1.42$. At 34.2 F, $P_{wv} = 0.193$, and $h_{fg} = 1073.9$.

Calculations: As a first step, find the *heat of vaporization* by substituting the experimental data in Equation A-1 as follows:

$$\begin{aligned} q_e &= (av + b)\,(P_{wv} - P_{av})h_{fg} \\ &= (0.0201 \times 2.1 + 0.055)\,(0.193 - 0.142)\,1073.9 \\ &= 5.32 \text{ Btu per (hr) (sq ft)} \end{aligned}$$

Second, determine the *sensible heat gain of the snow* by substitution in Equation A-2, thus:

$$\begin{aligned} q_s &= 2.6s\,(t_t - t_a), \text{ with } t_t \text{ equal to } t_p \\ &= 2.6 \times 0.048\,(34.2 - 29.2) \\ &= 0.62 \text{ Btu per (hr) (sq ft)} \end{aligned}$$

Then, apply Equation A-3 and find the *heat of fusion of snow* to be:

$$q_m = 746s$$
$$= 746 \times 0.048$$
$$= 35.8 \text{ Btu per (hr) (sq ft)}$$

Fourth, by substituting in Equation A-4 evaluate the *Heat Transfer by Radiation and Convection, viz.*:

$$q_h = 11.4\ (av + b)\ (t_f - t_a)$$
$$= 11.4\ (0.0201 \times 2.1 + 0.055)\ (34.2 - 29.2)$$
$$= 5.53 \text{ Btu per (hr) (sq ft)}$$

Now keep in mind that the total heat *output*, q_t, per unit area from the test data is $q_i - q_l = 43.7 - 1.42 = 42.3$, and the free area ratio, A_r, can be evaluated from

$$A_r = [q_t - (q_s + q_m)]/[q_o - (q_s + q_m)] = [q_t - (q_s + q_m)]/(q_o + q_h)$$

by substituting the several values as follows:

$$A_r = [42.3 - (0.62 + 35.8)]/(5.32 + 5.53) = 0.54$$

DISCUSSION

L. F. SCHUTRUM, Cleveland, Ohio: In Fig. 5, which gives the effect of wind velocity upon idling requirements, the measured wind velocities are relatively low. Were the wind velocity measurements made near the ground or near the test panels on the roof? Secondly, do the idling requirements given in this figure include the radiation component?

W. S. HARRIS, Urbana, Illinois: The authors are to be congratulated for supplying information in an area that is relatively new in our field and an area in which design and operating data are certainly not plentiful.

I have one question on which I would like to have the authors' comments.

The self-compensating feature of the free area concept as presented was based upon a condition in which the pattern of snow was more or less uniform over the entire panel. That is, there would be snowflakes and openings where the walk showed through very close together over the entire walk. Would this same self-balancing feature apply where you have the snow melted in streaks?

A conventional snow melting system may have pipes on 12 in. centers, and if it doesn't keep up with the load, there is likely to be one area of the walk that is completely free of snow while snow still remains between the coils. Would the self-compensating feature apply in this case, or would the reduction in resistance to heat flow from the pipe directly over the pipe to the air be so reduced that a greater proportion of the total heat flow would occur in that path and thus still further retard the melting of the snow between the pipes?

W. J. MAYER, Baltimore, Md.: I have read your previous articles and they are mighty interesting. I feel that it is good to have someone get into such analytical work to break down these things so that the basic principles and resulting effects can be more readily understood. However, when you get together with designing engineers, a great many of them will say, "We don't get paid for doing all of those calculations. We would like to see them boiled down into a few simple figures." I am wondering if Mr. Chapman has attempted to correlate his work to get it down into simple figures, let us say 100 Btu or 200 Btu, whatever it may be, for a certain snow-melting condition, so that they can be very simply applied by the engineer.

The next thing would be about carry-over, and the flywheel effect in heating so as to be ready for a snowfall. If ever you have had to calculate snow-melting for an airport air strip, you will find that the standby charges would be tremendous to keep the un-

covered strip warm. In the usual snow-melting jobs the owners are prone not to warm the slab before it snows, for economy reasons, or because they do not know in advance it will snow.

AUTHORS' CLOSURE (Mr. Katunich): Concerning Mr. Schutrum's question on wind velocity—you realize that these are actual outside wind velocity data over which we had no control. Inability to obtain higher wind velocities was a drawback. Wind velocity measurements were made on the roof of the laboratory near the snow melting apparatus. Eight or nine miles per hour was the highest wind velocity obtained in any test run.

The methods used to adjust for radiation effects are explained in the paper. Briefly, we corrected for radiation effects in one of two ways. The first method was to run the tests on days when the unheated panel temperature was exactly equal to the air temperature. The second method was to supply enough heat to a standard panel on cloudy days to raise it to air temperature; and then, in calculating idling requirements, we would subtract the heat supplied to the standard panel from the other test panels.

I would like to add that the equation stated in the paper, however, does include the radiation component. When we made the measurement we measured it during conditions, as pointed out, so that we were getting the true convection from the panel as indicated in the curve.

AUTHOR'S CLOSURE (Mr. Chapman): The self-balancing factor mentioned comes in automatically if the system tends to fall behind. If the pipes are spaced so that the area above the pipes is free of snow, but there is some accumulation of snow between the pipes as mentioned by Professor Harris, then in that area between the pipes we have the self-balancing effect.

The difference between a conventional system that is installed in a walk or a driveway and our laboratory system is that the laboratory system had an isothermal surface whereas the conventional system may not have an isothermal surface immediately after start up. We had to control as many variables as possible so we had a truly isothermal plane to work with. That accounts for the neat, uniform distribution that you saw in the illustration. A conventional system, using pipes, that has not been turned on sufficiently far in advance will give a *striped* effect during the transient period before equilibrium is established.

If, however, the striping occurs after the system has been in operation, then we may assume that the heating plant is inadequate. In either case, the self-compensating feature of the free area concept would hold true. If snow lies on the surface, mass and heat transfer are reduced, hence all energy reaching the surface will warm and melt snow.

Professor Harris raised the question of thermal resistance. True, there is less thermal resistance in the path from the pipe to the surface directly over the pipe than to the surface between the pipe. That explains how stripes occur. Let us assume we have stripes.

The surface temperature of the clear area must be above freezing; whereas, the temperature of the covered area must be below freezing. There will, therefore, be a lateral transfer of heat that will tend to eliminate the stripes.

Both of the points covered by Mr. Mayer are very well taken. The first one, which is a quick way of using the information given in the paper, I believe is covered in Table 2. We have 4 values of air temperature, 3 values for wind velocity, 3 rates of snowfall, and 2 values for relative humidity. Values are given for output and for fluid temperature for a system that is completely free of snow (a free area ratio of unity), and for a system that is completely covered (a free area ratio of zero).

It is certainly true that engineers do not have time to go through a complete analysis for a typical job. Occasionally a job might be big enough, but as a rule the engineer is pushed rather hard. Table 2 was prepared for those cases and contains the solutions to the required equations.

The second question was one of idling load. In certain areas, Boise, Idaho, for example, which is typical of the western slope of the Rocky Mountains, the idling requirements might be 40 times as great as the melting requirements. In other words, to maintain a temperature of 32 F on the surface for all of the hours that the air temperature was below freezing, but no snow was falling, forty times as much fuel would be required for idling than for melting the snow that does fall. Idling generally would be prohibitive.

If the owner is buying steam, for example from a steam company, it is possible to return the condensate at some lower temperature and subcool the condensate to say 160 F. The quantity of condensate may be sufficient to handle the idling load.

Essentially, then, you are getting free fuel for the snow melting system because rates are based on quantity of condensate and are independent of temperature. In a residential job, however, or a system that is designed primarily for melting snow, it is not likely that you can afford the luxury of maintaining 32 F all winter long.

1578

No. 1578

SMALL-PIPE PERIMETER HEATING IN A RESIDENCE†

By D. R. BAHNFLETH*, URBANA, ILL., H. T. GILKEY**, CLEVELAND, OHIO, AND C. F. CHEN††, PROVIDENCE, R. I.

INVESTIGATIONS of warm-air heating systems have been conducted in Warm Air Heating Research Residence No. 2 since 1947. These investigations have included studies of a high sidewall delivery system and a warm-air ceiling panel system[1], the introduction of outdoor air into the system and furnace-closet ventilation[2], and small-pipe perimeter systems.

The success of the warm-air perimeter system in homes with concrete slab floors led to the use of the perimeter concept for heating homes built over basements and crawl spaces. A preliminary study in Research Residence No. 2 during the latter part of the 1951–52 heating season resulted in the research program begun in the fall of 1952. This paper reports in part the results obtained during the 1952–53 and 1953–54 heating seasons.

The principal objectives were to determine:

1. The degree of uniformity of the room-air temperatures maintained by a perimeter heating system.
2. The effect of heat addition to the basement on both the room- and basement-air conditions and on the operating costs.
3. The operating characteristics of a small-pipe duct system and furnace; and
4. To compare the performance of a perimeter system with that of a high sidewall convection system.

DESCRIPTION OF RESIDENCE

The Residence, shown in Fig. 1, is a one-story frame structure with full basement. Walls are insulated with 3⅝ in. mineral wool batt type insulation with vapor barrier attached. Ceiling is insulated with 5 in. of mineral wool. The

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† This investigation was part of the cooperative project jointly sponsored by the Engineering Experiment Station of the University of Illinois and the *National Warm Air Heating and Air Conditioning Association.*

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[1] Exponent numerals refer to References.

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calculated coefficient of heat transmission, U, for the walls and ceiling is 0.07 Btu per (hr) (sq ft) (F deg). Windows, with the exception of the double glazed fixed window in the living room, are of the horizontal sliding type with storm sash attached to the window sash. Doors are of conventional wood and glass construction and are equipped with storm doors. The windows are weather-stripped. Usable floor area is about 1000 sq ft. Room dimensions and arrangement are shown in Fig. 2.

The heated space consisted of the first story rooms as well as the entire basement. The design heat loss[3], for −10 F outdoors and 70 F indoors, was 52,300 Btuh.

FIG. 1—WARM AIR HEATING RESEARCH RESIDENCE NO. 2

TABLE 1—FUEL-INPUT RATES, AIR-FLOW RATES, AND HEAT-INPUT RATES TO THE FIRST STORY AND BASEMENT

SERIES	FUEL-INPUT RATE, CU FT PER HR	TOTAL AIR-FLOW RATE, CFM[a]	HEAT-INPUT RATE, BTUH	
			BASEMENT	FIRST STORY
F-11	46.0[b]	340	0	31,000
F-21	51.4[c]	385	10,000[d]	31,000
F-24	65.5[c]	490	21,300[e]	31,000

[a] Computed air-flow rate for 100 F temperature rise through furnace.
[b] Input equals first-story loss divided by product of assumed bonnet and duct transmission efficiencies, 80 and 85 percent, respectively, for natural gas at 1000 Btu per cu ft.
[c] Input determined by same method as above, with an assumed duct transmission efficiency of 100 percent.
[d] Equivalent to the calculated above-ground heat loss.
[e] Equivalent to the total calculated heat loss.

This total consisted of 21,300 Btuh for the basement and 31,000 Btuh for the first story.

FURNACE AND DUCT SYSTEMS

The duct systems were designed in accordance with *Manual No. 10*[4] of the *National Warm Air Heating and Air Conditioning Association*. The 4-in. diameter ducts were connected directly to the bonnet of a gas-fired furnace with rated input of 70,000 Btuh. The supply outlets, located at the outside wall under the

windows, were all 2¼ x 14-in. floor diffusers. Each supply duct was equipped with a volume damper with which the system was balanced. The diffuser valves were fully opened at all times.

The 2 duct systems used are shown in Fig. 3. In Series F-11, 8 outlets were used to supply warm air to the first-story rooms. In Series F-21 and F-24, 6 outlets were used on the first-story and 3 outlets were used in the basement. The three 2¼ x 14-in. floor diffusers used to supply heat to the basement were located near the underside of the floor joists and used as ceiling diffusers.

One centrally located return-air grille was located on the first story. It consisted of a 24 x 6-in. baseboard grille and a 24 x 8-in. floor grille combination and was used

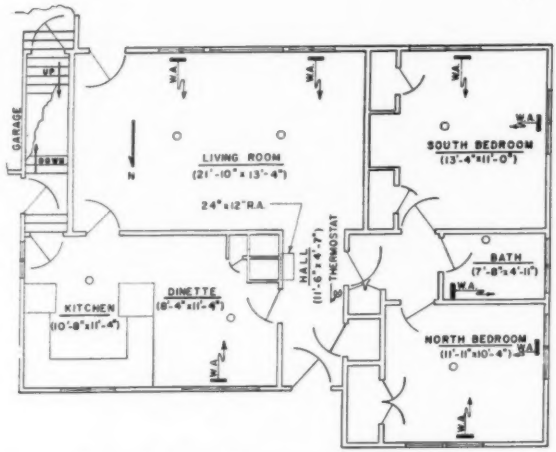


FIG. 2—FLOOR PLAN OF RESIDENCE. ALL WARM AIR OUTLETS ARE 2¼-IN. X 14-IN. FLOOR DIFFUSERS. R.A. SHOWS RETURN AIR OPENING, W.A. SHOWS WARM AIR OPENINGS; CIRCLES SHOW LOCATIONS OF THERMOCOUPLE STANDARDS

for all studies. The total free area of the grille combination was 289 sq in. The 24 x 12-in. opening in the floor was reduced to 12 x 12 in. with a reducing elbow, and the return duct was of constant size from the elbow to the return plenum.

A 9-in. diameter duct was used for the basement return in the studies conducted with heat added to the basement. The intake to the basement return was located approximately 1 ft below the floor joists during most studies. A short test was made with the basement return in a horizontal position so that the intake was at the basement floor level. Each return-air duct was equipped with a volume damper.

INSTRUMENTATION

Approximately 210 copper-constantan thermocouples were provided for temperature measurement. Room-air temperatures were measured at 4 levels at the locations shown in Fig. 2. Instrumentation was also provided to measure indoor rela-

tive humidity, electrical inputs to the lights and appliances and to the blower, the fuel input rate, the time of burner and blower operation, the air-flow rates in the duct systems, and the pressure losses of the duct systems.

EXPERIMENTAL PROCEDURE

Air-flow rates through the furnace, fuel input rates, and heat input rates to the basement and first story are shown for each study in Table 1. The furnace was

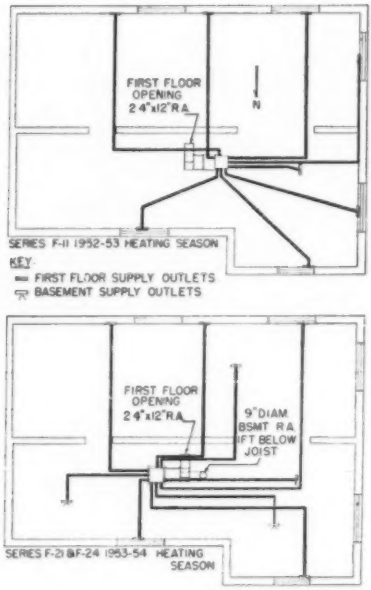

FIRST FLOOR OPENING 24"x12" R.A.
N
SERIES F-11 1952-53 HEATING SEASON
KEY
FIRST FLOOR SUPPLY OUTLETS
BASEMENT SUPPLY OUTLETS
FIRST FLOOR OPENING 24"x12" R.A.
9" DIAM BSMT R.A. 1FT BELOW JOIST
SERIES F-21 & F-24 1953-54 HEATING SEASON

FIG. 3—BASEMENT PLAN AND DUCT LAYOUTS

adjusted for continuous air circulation[5] for each study. The fuel input rate for each study was determined from the calculated heat loss. The desired input was obtained by changing the burner orifice and by adjusting the burner manifold pressure at the pressure regulator of the furnace. The air-flow rate through the furnace was set to give an air temperature rise in the furnace of 100 deg with the burner and blower operating continuously.

The procedure used to obtain the desired heat inputs to the basement was as follows:

1. Fuel input rate was set for Series F-21 based upon the first-story heat loss (31,000 Btuh), and the above-ground basement heat loss (10,000 Btuh). The total air-flow

rates to the first story and basement were set near values calculated for a 100 deg rise with a bonnet static pressure of approximately 0.190 in. water gage.

2. Individual air-flow rates in the first-story runs were adjusted to obtain satisfactory room-to-room temperature balance while maintaining the same total air-flow rate to the first story.

3. Temperature rise through the furnace was then measured and the basement air-flow rate was adjusted until a rise of 100 deg was obtained. During these adjustments the bonnet static pressure was maintained constant.

4. In succeeding tests the fuel input rate was adjusted to the desired value. Then only the blower speed and basement dampers were adjusted until a rise of 100 deg was obtained with a bonnet static pressure of 0.190 in. water gage.

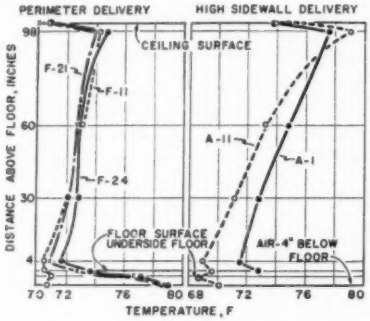


FIG. 4—ROOM-AIR TEMPERATURE GRADIENTS IN LIVING ROOM: WITH OUTDOOR TEMPERATURES FROM 31–37 F; AT NIGHT WITHOUT SUN; WITH CONTINUOUS BLOWER OPERATION; WIND VELOCITY OF 3 TO 7 MPH, SOUTHWEST

5. Air-flow rates in the returns were adjusted by means of dampers so that the return-air flow from each zone equalled that which was delivered to the zone.

The thermostat, located at the 30-in. level in the entrance hall, was set to maintain a temperature of 72 F. The thermostat differential was adjusted to cause burner on-periods of 3 to 4 min. when the outdoor temperature was between 20 and 30 F.

The Residence was occupied by a family of two adults during Series F-11. During Series F-21 and F-24 the Residence was furnished but unoccupied.

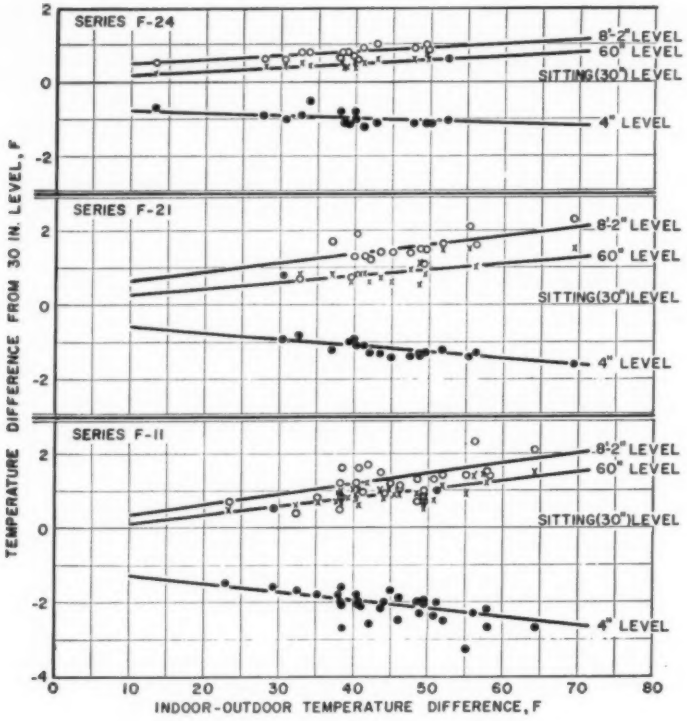
Comfort was evaluated on the basis of the air and surface temperatures in the Residence. These temperatures were recorded at 6:00 a.m. in order to minimize the effects of occupancy and solar radiation. The results are all based on these 6:00 a.m. readings unless otherwise noted.

RESULTS

Uniformity of Room-Air Temperatures: The cyclical variation of room-air temperatures at the 30-in. level was small, less than 1 deg, with all 3 series. The re-

sponse was considered satisfactory since no lag or overrun was experienced during periods of rapidly changing outdoor temperature.

The room-to-room temperature balance at the sitting level was satisfactory for each of the systems studied. In general, the temperature difference between rooms



	TEMPERATURE DIFFERENTIALS, F									
	INDOOR-OUTDOOR TEMPERATURE DIFFERENCE									
	35 F					70 F				
	F-11	F-21	F-24	A-11	A-1	F-11	F-21	F-24	A-11	A-1
3°FI-30°FI	1.8	1.0	0.9	1.8	0.9	2.6	1.6	1.2	2.6	1.7
3°FI-60°FI	2.5	1.7	1.3	3.8	2.3	4.1	2.9	2.0	6.9	4.7
3°FI-3°CLG	2.9	2.3	1.7	8.1	5.4	4.6	3.7	2.3	16.9	12.4

FIG. 5—AVERAGE ROOM-AIR TEMPERATURE DIFFERENTIALS

was less than 2 deg. The maximum difference in temperature between rooms for each of the studies conducted was 2.4, 1.8, and 2.3 deg, for Series F-11, F-21, and F-24, respectively. This maximum unbalance occurred during days when high wind velocities and low outdoor temperatures were experienced. The lowest room-

air temperatures during Series F-21 and F-24 occurred in the bedrooms. The capacities of the single runs to each of these rooms were less than the design heat losses, but the installation was made in this manner to determine if heat addition to the basement would have any material effect on the balance. Heat addition to the basement did not improve the balance in the first-story rooms.

The room-air temperature gradients in the living room are shown in Fig. 4 for both the perimeter systems and the high sidewall systems studied previously[1]. The gradients in the left-hand portion of the figure are for the 3 conditions of heat addition to the basement with the perimeter system; the gradients on the right are for 2 conditions of heat addition to the basement with the high sidewall system. In Series A-1 the basement was heated; in Series A-11 the basement was unheated. These can be compared with Series F-24 and F-11, respectively.

Several observations which apply to conditions in all rooms can be made. The addition of heat to the basement did not materially improve the gradients experienced with the perimeter system. Since the gradients with the basement unheated, Series F-11, were very small, only small improvements due to heating the basement could be anticipated. The addition of heat to the basement with the high sidewall system did improve the first-story gradients as is shown in Fig. 4. The improvement was due to the panel heating obtained from the warm floor. The gradients of the perimeter system were more satisfactory than those experienced with the high sidewall system.

The effect of outdoor temperature on the average room-air temperature differentials for the perimeter systems is shown in Fig. 5. The tabulation at the bottom of Fig. 5 summarizes the average differentials for the 3 perimeter systems and the 2 high sidewall systems. Even though the effect of heat addition was slight in the case of the perimeter systems, significant improvement was experienced in the temperature differences from the floor to the sitting level. As will be shown later this can be attributed to the higher floor-surface temperatures maintained when heat was added at the basement.

The temperature differentials in the living zone (floor level to breathing level) which were experienced with the perimeter system were more satisfactory than those experienced with the high sidewall system. From the tabulation in Fig. 5, it can be seen that the temperature difference from the floor to the breathing level with Series F-24 was 2.0 deg and that for Series A-1, it was 4.7 deg when the outdoor temperature was 0 F (indoor-outdoor temperature difference = 70 deg), and the basement was heated. When the basement was unheated, the temperature differences in the living zone were 4.1 and 6.9 deg in Series F-11 and A-11, respectively. The improvement in the average living zone differentials caused by heating the basement was approximately the same for both systems investigated.

The temperature differentials from the floor level to the ceiling level with the perimeter systems were smaller than those experienced with the high sidewall systems. As is shown in the table these differentials were 2.3 and 12.4 deg for Series F-24 and A-1, respectively, at an outdoor temperature of 0 F. When the basement was unheated, these differentials were 4.6 and 16.9 deg, in Series F-11 and A-11, respectively. The large floor to ceiling differentials observed for the high sidewall systems (A-1, A-11) do not indicate an uncomfortable condition since the major portion of this temperature difference occurred between the breathing and ceiling levels. They do indicate, however, that the heat loss of the Residence may be greater when heating with the high sidewall system.

A comparison of the floor to ceiling temperature differentials and the gradients (Fig. 5) experienced with the perimeter and high sidewall systems, indicates that the average room-air temperature was lower with the perimeter systems. These

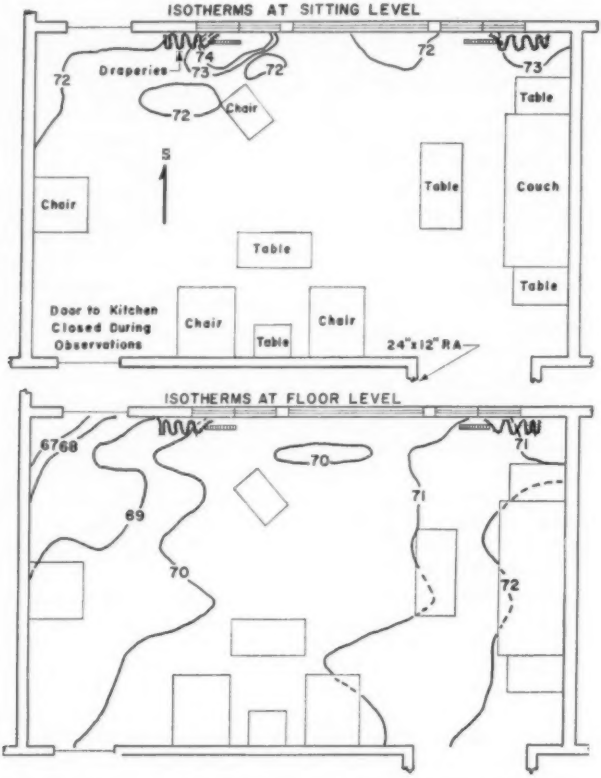


FIG. 6—ISOTHERMAL CONTOURS IN LIVING ROOM: WITH OUTDOOR TEMPERATURE AT 14 F; AT NIGHT WITHOUT SUN; CONTINUOUS BLOWER OPERATION; WIND VELOCITY AT 4 MPH, NORTHWEST

lower average room-air temperatures reduced the heat loss of the Residence, thus resulting in a lower operating cost. Since the reduction in room-air temperature was greatest at the ceiling level, amounting to approximately 10 deg, the ceiling losses were appreciably reduced.

Special studies were conducted during Series F-21 and F-24 to determine the uniformity of room-air temperature at the floor- and sitting-levels in the living room.

Air temperatures at the 2 levels were measured at 76 locations by portable thermocouples. Since the results of all of these special studies were similar, only those obtained on March 4, 1954 are presented in Fig. 6.

The room-air isotherms at the sitting level ranged from 71.3 F near the picture window to 74.5 F above the floor diffusers, and most sitting-level air temperatures were between 72 and 73 F.

The isotherms at the floor level ranged from 67.0 F at the southeast corner to 72.1 F at the northwest corner. About one-fourth of the total area at the floor level was below 70 F. This cool area was caused by the infiltration from the outside door and the cold down drafts from the door and east wall which is partially exposed to the attic. The pattern of the isotherms at the floor level indicates that the

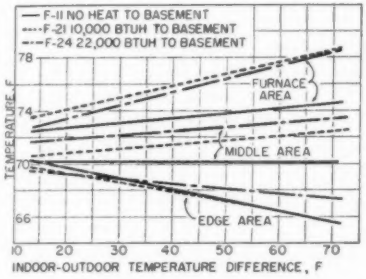


FIG. 7—EFFECT OF OUTDOOR AIR TEMPERATURE AND HEAT INPUT TO BASEMENT ON SELECTED FLOOR SURFACE TEMPERATURES

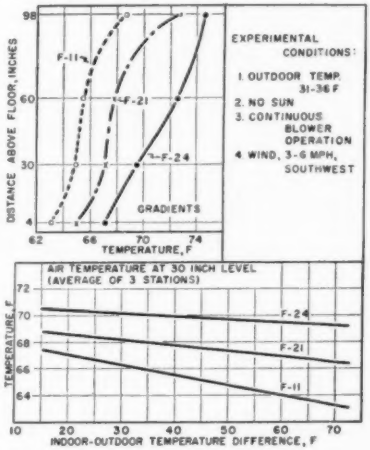


FIG. 8 (*Right*)—AIR-TEMPERATURE GRADIENTS AND AIR TEMPERATURES IN BASEMENT

warm air from the floor diffuser had little influence on the down drafts at the exposed corner.

Floor-Surface Temperatures: The floor-surface temperature at any point depended upon its location with respect to the furnace and exposed walls. In the immediate vicinity of the furnace, the floor-surface temperatures were satisfactory when the basement was unheated and tended to increase with decreasing outdoor temperatures. Fig. 7 shows that the addition of heat to the basement served only to increase these temperatures further, thus increasing the heat regains to this area. The middle area of Fig. 7 was influenced by the heat losses from the ducts but was not greatly affected by the exposed walls. The addition of heat to the basement increased these floor-surface temperatures, but little effect of outdoor temperature was apparent. Since this is both the largest area considered and that most used by the occupants, the addition of heat caused a significant improvement.

The floor-surface temperatures of the edge area, near the perimeter of the Residence, were primarily affected by the exposed walls. The lower curves of Fig. 7

show that the floor-surface temperature in the exposed corner of the south bedroom was not increased appreciably during Series F-21, when the amount of heat added to the basement was equal to its above-ground heat losses. A small increase in temperature was observed when the basement was fully heated in Series F-24. The improvement at the perimeter in Series F-24 was attributed to the higher register-air velocities in the basement with the higher heat inputs. Consequently, the air distribution in the basement was improved and some of the warm air from the outlet reached the exposed corner.

In view of the floor-surface temperatures experienced during these studies, the most desirable location for basement outlets would appear to be in the vicinity of the exposed corners of the Residence. Because of the duct and bonnet losses, the

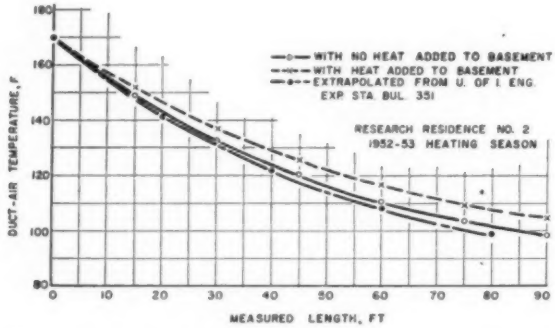


FIG. 9—AVERAGE TEMPERATURE DROP IN 4-IN. DIAMETER DUCTS FOR VELOCITY OF 600 FPM

floor-surface temperatures in the central area of the house will be satisfactory without the addition of heat to the basement in this area.

Basement-Air Temperature Variation: The air-temperature gradients in the southwest section of the basement are shown in the upper portion of Fig. 8. Although the gradients were not improved by heat addition to the basement, the air-temperatures at each level were improved by the addition of heat.

The effect of outdoor temperature on the air temperatures at the sitting level in the basement is shown in Fig. 8. A comparison of Series F-11 and F-24 shows that the air temperature at the 30-in. level in the basement increased from 63.2 F to 69.2 F at an outdoor temperature of 0 F when the basement was fully heated. Locating the basement return at the floor level caused a further improvement of about 1 deg in the air temperatures at the floor and sitting levels.

Pressure Losses: The observed pressure losses for the study in which no heat was added to the basement (Series F-11) were 0.107 in. water gage on the warm-air side and 0.016 in. water gage on the return side at an air-flow rate of 340 cfm. As was mentioned previously, the pressure losses on the warm-air side were adjusted to 0.190 in. water gage during Series F-21 and F-24 in which the flow rates were 385 and 490 cfm, respectively. The return losses were 0.015 and 0.042 in. water gage for Series F-21 and F-24, respectively.

Temperature Drop in 4-in. Diameter Ducts: Studies were made during the 1952–53 heating season to determine the temperature drop in 4-in. diameter ducts. The duct system was that used for Series F-11, and the individual runs were of various lengths and included different numbers of 4-in. adjustable elbows. The air-flow rate in the ducts, and the temperature of the entering air were varied to determine the effect of velocity and entering air temperature on the temperature drop. The results of these studies have been compared to temperature drop information obtained by extrapolation from previously published data[6].

Fig. 9 summarizes the results of these studies for duct-air velocities of 600 fpm and for 2 conditions of heat addition to the basement. The upper curve shows the average temperature drop when 10,000 Btuh was added to the basement to offset the above-ground heat losses. The middle curve shows the average temperature drop for the unheated basement condition, and the lower curve is that obtained from

TABLE 2—EFFECT OF HEAT ADDITION TO BASEMENT ON TOTAL ENERGY INPUT TO THE RESIDENCE FOR THREE PERIMETER AND TWO HIGH SIDEWALL SYSTEMS

(*Indoor-Outdoor Temperature Difference* = 80 *F*)

SERIES	TYPE OF SYSTEM	HEAT INPUT TO BASEMENT, BTUH	TOTAL ENERGY INPUT TO RESIDENCE	
			BTU PER DAY	INCREASE DUE TO HEATING BASEMENT, PERCENT
F-11	Perimeter	0	1,070,000	—
F-21	Perimeter	10,000	1,200,000	12.1
F-24	Perimeter	21,300	1,200,000	12.1
A-11	High Sidewall	0	1,250,000	—
A-1	High Sidewall	26,285	1,380,000	10.4

the results of previous work. Good correlation was obtained, especially for duct lengths which would commonly be found in small-duct heating systems. The addition of heat to the basement caused a slight decrease in the temperature drop.

Overall duct losses were measured and were found to be approximately 15 percent of the input to the furnace. Even though the transmission efficiencies were not affected by the addition of heat to the basement, the magnitude of the duct losses was smaller when the basement was heated. This occurred because no allowance for transmission loss was made when heat was added to the basement. Thus, for a given register delivery and air temperature entering the duct, the air-flow rate will be less if no transmission loss is assumed to occur.

Fuel Consumption: During 2 of the 3 studies reported, the Residence was not occupied. The energy lost from the structure during any day equalled that supplied by the furnace plus the small amount required to operate the instruments and blower, and to maintain the water heater control temperature. When the Residence was occupied, the increased use of the water heater, the lights, and the kitchen range supplied more of the energy required and resulted in lower daily fuel consumption. A realistic comparison of the cost of heating can be obtained only by considering total energy input during each of the studies.

In Table 2 the total energy input to the Residence in Btu per day is shown for the local design temperature, −10 F. The data were taken from curves of total energy input plotted as a function of indoor-outdoor temperature difference for each of the studies conducted. Values from these curves represent averages for the range of weather conditions experienced, and it was assumed that the percentage of the total input used to offset the heat loss was constant regardless of the source of the energy. Adding heat to the basement caused an increase of 12 percent in fuel consumption whether heat was added to the basement at the rate of 10,000 or 21,000 Btuh. In each case the additional fuel consumption was less than would be predicted from the calculated heat loss of the basement. This can be attributed to the increased panel gains to the first story as the heat addition to the basement was increased. Although the panel gains could not be measured, they were evidenced by a decrease in the average length of burner cycle as the rate of heat addition to the basement was increased.

It has previously been shown[1] for the high sidewall system that, for the same condition of occupancy, the furnace consumed approximately 10 percent more fuel when the basement was heated. As is shown in Table 2 this percentage increase was also experienced in the total energy input to the Residence. Under similar conditions of heat addition to the basement the total energy consumption was less for the perimeter system.

CONCLUSIONS

Cyclical variation in room-air temperature was small with the three perimeter systems studied. The room-to-room temperature balance was not affected by heat addition to the basement. This was especially noticeable in the bedrooms where the duct capacities were less than the calculated heat losses of the rooms.

Only small improvements in the room-air temperature gradients and differentials were observed when heat was added to the basement. Since the differentials experienced with most perimeter systems are good, only small improvements were anticipated.

The floor-surface temperatures were satisfactory without heat addition to the basement except near the exterior walls. There was, however, an increase in the floor-surface temperatures in the region between the perimeter and the area directly affected by the furnace bonnet when heat was added to the basement. The floor-surface temperatures at the perimeter were improved very little by adding heat to the basement. It appears that the basement outlets should be located near the exposed corners of the Residence.

The total energy input increased 12 percent when the basement was heated. This percentage increase is much less than that which would be predicted from the calculated heat loss of the basement. Previous investigations of a high sidewall system indicated a 10 percent increase in energy input was needed when the basement was heated.

In general, the temperature differentials and gradients with the perimeter systems were better than those experienced with the high sidewall system under similar conditions of operation. The living zone differentials differed by only 1 to 3 deg, depending upon the outdoor temperature, but a greater difference was observed in the floor-to-ceiling temperature differentials. Even though this is not important from the standpoint of comfort, it does indicate that a lower heat loss and conse-

quently a lower operating cost could be anticipated for the perimeter system. A comparison of the total energy input for the perimeter and high sidewall systems shows that for similar conditions the total energy consumption was less for the perimeter system.

Good agreement was found between measured temperature drop in 4-in. diameter ducts and temperature drop information published previously[6]. Some reduction in duct heat loss occurred when the basement was heated. In duct lengths which would normally be found in perimeter systems, this reduction was small.

Overall duct losses were found to be 15 percent of the input to the furnace. This loss is in good agreement with the assumed duct transmission efficiency of 85 percent commonly used.

ACKNOWLEDGMENTS

It is expected that these results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is hereby made to the manufacturers who cooperated by furnishing materials and equipment used in this investigation.

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DISCUSSION

S. F. GILMAN, Syracuse, N. Y.: The temperature gradient plot in Fig. 4 certainly shows that perimeter delivery of the supply air was definitely superior to high sidewall delivery. However, I question that the high sidewall system has been operated under equally favorable circumstances. As I recall it, this residence utilized about six 12 inch by 6 inch registers. Considering a typical outlet to have 65 percent free area, the total free area amounts to just about two square feet.

Since the furnace was adjusted to obtain a 100 deg F rise, which is the *NWAHACA* recommended procedure, the resultant air flow rate is very low. Table 1 shows it to be 340 cfm for Test F-11. As a consequence, the average discharge velocity from the high sidewall registers was about 170 fpm. With 165 F at the bonnet, the average discharge temperature was undoubtedly high—perhaps 140 F or more. Therefore, a slow moving stream of hot air is being delivered to the room just a foot or so beneath the ceiling. This can only lead to the poor performance indicated in Fig. 4.

My principal disagreement is with the use of the 100 deg F rise concept with high sidewall distribution systems. In the 1953 and 1954 Transactions are papers on room air distribution. These point out that high sidewall performance is significantly improved by either increasing the discharge velocity or decreasing the supply air temperature. The research was also conducted at Illinois but sponsored by the American Gas Association.

There is little doubt about the effectiveness of floor diffusers. I do believe, however, that the temperature curves of Fig. 4 would be much closer together under operating conditions that are more logical for high sidewall systems, such as 400 fpm discharge velocities and a furnace temperature rise of 70 deg F.

I note that the total heat energy requirement is considerably greater for the high sidewall system. It is not mentioned, however, that the principle of the floor diffuser is to *blanket* the outside walls with warm air. It would seem that this would tend to bring the fuel consumptions of the 2 systems more into agreement. Do the authors have data on wall surface temperatures at points behind the diffuser air pattern? What is their opinion concerning the relative fuel consumptions of a perimeter system versus floor registers located along inside walls?

Fig. 7 shows that floor surface temperatures in the edge area tend to be low and are difficult to raise. Analysis shows that there is an easy thermal path from the edge floor material through the outside wall to outdoors. It appears that a cold floor near the periphery is a problem that might be solved by a better thermal barrier between the floor and wall. The authors are asked to describe the floor joist and outside wall configuration, because, as I recall, it is not conventional wood construction. Perhaps they can suggest other ways of improving surface temperatures along the edge area.

Finally, since this basement has an unusual amount of glass, do the authors feel that 5 or 7 percent increased fuel consumption is reasonable for general application engineering data?

P. R. ACHENBACH, Washington, D. C.: I am very much interested in this paper. I would like to ask a question and take a little pot shot at the description of this system as a perimeter heating system.

I would like to ask the authors whether they or the *National Warm Air Heating and Air-Conditioning Association* have actually defined perimeter heating. I recall a few years ago when this idea was first developed that it incorporated a duct around the perimeter of a house imbedded in concrete, or otherwise, whereas this system is not much different from the conventional forced warm air system except that perhaps the registers are extended to the outside walls, I believe it would be helpful to the industry and the public to have a clear-cut definition of what perimeter heating is. It seems a little bit of a misnomer to me for this system.

Then I would like to say that there should be some caution used in taking these results and applying them to houses that are less thoroughly insulated than this one. The authors point out that the walls and ceiling have 4 and 5 inches of mineral wool. I believe the doors and movable windows are weather-stripped and they have storm windows on the windows.

Our experience at the Bureau of Standards with a house that had no insulation in the walls and usually no storm windows was that the temperature differences between rooms and between levels are much greater, and generally speaking that these temperature differences are proportional to the amount of heat which must be put into the space.

When you insulate a structure thoroughly, the amount of heat necessary to discharge into the space is correspondingly smaller and it takes less temperature head between the areas to derive the heat with an air stream or any other method by which heat is transferred from place to place.

I believe some of the conclusions that are drawn in the paper could not be made quite so flatly in an uninsulated house, and especially with regard to the differences in temperature.

I would also like to comment on the heat loss in the basement. I believe where they did not deliver heat to the basement registers they considered it no heat in the basement.

On the other hand, I believe the duct loss was of the order of 7,000 to 10,000 Btuh, and that, in many houses, is sufficient to raise the temperature to, at least under the floor joists, where there is a heat flow upward instead of downward, or at least approximately that.

I think there is a question whether any delivery of heat is really necessary in this house with the perimeter system, as they call it, at all.

I would like to compliment the authors on the studies of isotherms at the floor level and at the 30-in. level. We did that at the Bureau of Standards for some space heater heating and it isn't nearly as favorable naturally as this system, but I think it does reveal better than a few spot temperatures how much of the room is livable. After all, we are not only concerned in whether the center of the room is comfortable to sit in, but whether the chairs that are usually against the wall are also comfortable to sit in. So I think that this idea of investigating the isotherms over the whole floor area is very worthwhile.

I would like to question the expression of duct losses in terms of a percentage of the input to the furnace. At least in the wet heating industry, it is usually expressed in terms of a percentage of the calculated load of the house rather than the input to the heating system.

This is a very interesting paper and my mild criticism is not to say that I don't think it is very worthwhile.

I have another comment. I would like to sharpen up my comment a little bit about the wider usage or wider application of these data. I think perhaps more significant than the amount of insulation is the distinction between a house with a furnace in the basement and a basementless house. Our experience is that when the heating system is located in the basement, the loss from the furnace and the smokepipe and the ducts is usually enough to heat the air just under the joists, almost to the temperature above the floor. And I think that is supported in Fig. 4 where the basement was not heated. As they call it in this paper, the temperature below the floor joists was practically equal to the floor surface temperature. So that the floor was a neutral surface as far as heat transfer is concerned.

In an uninsulated frame house, the interior wall surface temperature at zero outside will be in the order of 58 F and the air falling down the wall will be cooled considerably. In a wall insulated as well as this, and with these air temperatures and doors I would expect the wall surface temperature was on the order of 70 F and so that the air moving down the wall couldn't be cooler at any lower than that, and that has a significant effect on the floor surface temperatures adjacent to the wall.

I wouldn't consider any of the floor surface temperatures that are shown in Fig. 6 to be uncomfortable. I think they are all warm enough for comfort. I would like to make one comment on why I think the perimeter system shows less heat loss than the ceiling outlet system. The wall diffusers could only affect the wall surface temperature in a relatively small area in the vicinity of each diffuser, and that is a fairly small percentage of the whole exterior wall whereas almost the entire ceiling would be elevated in temperature as a result of discharge of warm air by high wall outlets; so that I think the proportion of the 2 components that raised the temperature by the 2 systems is in favor of the perimeter system giving a lower overall heat loss.

R. B. RICE, Raleigh, N. C.: This paper has and will continue to stimulate a great deal of interest in engineering circles. The conclusions are definitely a step in the right direction. They should not be viewed as being final, however, as there is need for continued and additional research work along this line.

Fig. 2, the layout and floor plan of the building and the associated contributing factors which would influence the validity of the report and its explanation incites considerable curiosity. There are some 6 internal doors. The position of each or all will have

considerable influence on the heating load, the results, and specifically on the isotherms which are illustrated clearly in Fig. 6. I feel the authors should comment on this and should include a statement concerning these doors in their excellent paper.

Also, I notice the authors indicate the isotherms or the heat distribution only for Fig. 6 and for this one room which is 20 ft x 13 ft., the living room. This suggests that the other rooms have been disregarded from the standpoint of comfort. I feel this study certainly should be expanded to cover other spaces in the house.

Likewise, perhaps the authors might care to extrapolate the data to anticipate what might be the results with not quite as much insulation employed. In a building of this size and type it has been my experience that its insulation is many times neglected.

One other point: I think it would be excellent if the authors could comment pertaining to the transferring of this information for use by us in air-conditioning. Could this general data be used satisfactorily in warm weather cooling of a similar type of building?

T. J. BLACK, Detroit, Mich.: I am very interested in this report since I have just had occasion to read most of the Journal reports on the 4-in. duct system. I was called in as a consulting engineer for a city in Michigan which had trouble heating a builder's project of 80 homes all of which were unsatisfactory in heating.

In this report, 2 conditions were set up, one with 2 supply outlets in each of the corner bedrooms, and the other with the basement heated with one outlet in the bedrooms. It was the latter case which I proved unsatisfactory in heating. Have the authors any data on the above comparison?

In my calculations from the Journal's previous reports there seems to be an excessive pressure loss in supplying the cfm for a corner bedroom with one 4-in. duct. From the diagram it takes 2 or 3 elbows plus the butt takeoff which they show and the register elbow. I find that in my calculations it is impossible to heat an average corner bedroom with one 4-in supply. In other words, the resistance of the 4-in duct is so great that the air supply of 90 cfm cannot overcome the resistance of the duct. I would like a little explanation of the pressure losses for the one outlet in the bedroom and whether it totaled more than 0.20-in. WG.

E. R. TESKE, Chicago, Ill.: I presume that all combustion air was supplied by infiltration. I wonder if the authors have attempted to consider how the basement temperatures would be affected if they had to supply combustion air either through duct work or had to partition off the furnace, as they would in many municipalities that forbid air return from the space which includes the combustion equipment.

W. J. MAYER, Baltimore, Md.: What was the *U* factor of the outside wall?

AUTHOR'S CLOSURE, (H. T. Gilkey): Dr. Gilman's first comment involved the 100 deg F air temperature rise through the furnace and the use of the high sidewall diffusers. His implication was that high supply air temperatures would tend to increase the stratification particularly at the ceiling, thereby giving higher air temperature differentials and gradients and indicating perhaps less favorable performance for the inside wall delivery system.

What he says may very well be true, and yet the continuous air circulation concept, of which the 100 deg temperature rise is a part, was originally based upon research conducted in Warm Air Heating Research Residence No. 1 at the University of Illinois, and was confirmed in both Research Residences No. 2, with which this paper is concerned, and No. 3, which was a basementless house with a concrete slab floor placed on the ground.

In both Residence No. 1 and Residence No. 2, studies were conducted with high sidewall delivery systems, and better heating results, so far as uniformity and cyclical variations of room air temperatures were experienced using the continuous air circulation principle.

Dr. Gilman asked why, if perimeter diffusers blanket the wall with warm air, does the overall heat loss of the house apparently go below that which was found with higher-

air temperatures near the ceiling. We don't have the actual surface temperature information available. The fact is, though, that these perimeter diffusers do not actually scrub the outside wall. Furthermore, in looking at Fig. 6, the air isotherm drawing for the living room, the isothermals at the sitting level indicate there is quite a bit of entrainment of room air into the jets issuing from the floor diffusers and, as such, there was not a great blanket of high temperature air right next to the exterior wall. There is, shall we say, a blanket of more moderate air, and I don't believe the wall-surface temperature would be greatly increased by the use of perimeter diffusers.

We must also recognize that this house was thoroughly insulated with mineral wool in the walls and ceiling and that the windows were all double glazed.

He also mentioned something about the floor-surface temperatures near the outside wall, and asked about the construction of the house, particularly in regard to the connection of the floor joists with the exterior wall. The floor joists were open web steel beams, put in for potential test purposes. They did not join directly with the studdings in the exterior walls, but rather rested on steel plates on the top of the foundation wall. The stud spaces were insulated even down below the level of the first floor. However, the basement walls themselves were not insulated.

The question leads up to: How do we maintain higher floor-surface temperatures in the exterior areas of houses? In some houses, I think it would be impossible; gophers could get into some of the construction. In others, muskrats could get in. But it is a problem I think we will have to face up to. Extending the insulation down into the stud spaces between the floor joists will help. We may even come eventually to insulating the basement walls. I don't know.

Unfortunately, I am unable to speculate on the cost of heating a basement in a more conventional type of construction in which the basement windows are smaller. It might be less percentagewise, but anyone's guess is probably as good as mine.

Mr. Achenbach questioned whether or not there was a definition of perimeter heating. The *NWAHACA* considers warm air perimeter heating to be a warm air heating system in which perimeter diffusers are used and are placed at the outside wall and generally underneath a window. It is true that the term was initially applied to the system which we now call the perimeter loop system and is used in a basementless home having a concrete slab floor laid on the ground. The term, however, since about 1950 has been more generally applied to any warm air heating system in which perimeter diffusers are placed at the perimeter of the house. Perhaps that has not been made clear; nonetheless, I believe it is rather generally used that way.

Mr. Achenbach also warned against applying these results to less well insulated or to less insulated homes. The same comment was made by Mr. Rice. Similar results were obtained in Research Residence No. 3 with both perimeter-loop and perimeter-radial systems. In each of these systems the ducts were embedded in the concrete slab floor. In the case of the loop system, an embedded duct looped the entire perimeter of the house. In the case of the radial system the embedded ducts led directly from the furnace to the perimeter registers and diffusers.

Residence No. 3 had its walls uninsulated. There was no storm sash. The windows were not weather-stripped. The doors were equipped with storm sash or storm doors. The ceiling was insulated. Essentially the same results were obtained in Residence No. 3 as are recorded here for Residence No. 2 with, of course, a higher heat loss per square foot. Those studies have been previously reported in 2 papers to this Society.

Mr. Black asked for information on the air temperature conditions in the bedrooms In the first place, Fig. 4, to which he referred, applies only to the living room and does not apply to the bedrooms. The temperatures in the bedrooms using only one diffuser in each bedroom were lower than 72 F. This is mentioned specifically in the paper. The use of one diffuser plus introducing heat into the basement did not compensate for the one duct having a capacity less than the designed heat loss of the room.

So far as pressure losses in the duct system are concerned, they are spelled out in the paper. In any design, you are just going to have to take into account in your design

the heat loss of the area which you are considering, and the number of runs necessary to heat it. We cannot arbitrarily say that in any corner bedroom you must use 2 runs. You get much better air distribution if you have 2 diffusers, but if the room itself is small enough and your runs are of suitable length, you should be able to heat it satisfactorily with only one run. That is a matter of design.

Mr. Teske asked about the combustion air supply. In this case the furnace was located in the basement and so far as I know there are few, if any locations in which an outdoor air supply would be required by law. In some cases it might be very desirable in order to supply combustion air in that way. Information on that particular subject is given in University of Illinois Engineering Experiment Station Bulletin 427, which has to do with the introduction of both combustion and ventilation air in a warm air heating system.

Mr. Achenbach commented about the increased heat loss, using the inside wall delivery system. He mentioned that the ceiling air temperature had a larger exposure area involved than the increased air temperature near the outside wall with a perimeter system.

The *U* value of the exterior walls was 0.07, as figured by THE GUIDE, but in the heat loss calculations, Manual 3 of the *NWAHACA* was used and, in that publication, a minimum line was arbitrarily drawn at 0.09.

AUTHOR'S CLOSURE, (D. R. Bahnfleth): Another of Mr. Achenbach's questions referred to the heat addition to the basement. The 10,000 Btuh or the 21,000 Btuh added, was considered direct heat addition, something that could be taken into account. It is true that there was a duct loss, also a bonnet loss and a loss from the smokepipe in the basement. The fact that these losses were not sufficient to maintain floor-surface temperatures or underside floor-surface temperatures, which would indicate a heat flow to the first floor, was shown on gradients, and the floor-surface temperatures. We did improve the conditions by adding directly an additional amount of heat which was equivalent to the mentioned heat loss.

I will have to take the blame for expressing the duct loss in percent of input to the furnace. It should probably have been expressed on the capacity of the system or the input to the duct system which would increase the percentage duct loss to a little more than 15 percent.

Referring to the questions by Mr. Rice on the isotherms, he said that he felt we might be gilding the lily a little bit by showing only the isotherms in the living room. We chose the living room because we had traversed that room previously. We did not make as complete a traverse on the other rooms, but we have checked them and find the conditions are similar to those in the living room.

The effect of doors would be appreciable if the doors were closed and sealed. With the central return, it would then be essential that you either have a return grille in the door or a return in each room.

In our case the doors do not fit tightly at the bottom. We have a slot of about at least 1-in. which allows us to return air. However, all of our studies are made with the doors open so that they do not influence the results.

In the studies of floor isotherms, however, we did close the door between the kitchen and the living room. His point is very well taken. If a residence or the rooms are open and air is allowed to freely interchange between rooms, the balance would be quite different from what it would be if the doors were closed.

1579

No. 1579

HEAT LOAD CALCULATIONS BY THERMAL RESPONSE

By W. R. BRISKEN*, BLOOMFIELD, N. J., AND S. G. REQUE**, SCHENECTADY, N. Y.

DURING recent years great efforts have been made to calculate more accurately the heating and cooling load requirements of buildings. Electrical analogues have been used especially when storage capacities of building materials needed to be considered. Various authors have presented results of electrical analogue computer calculations. They have clearly demonstrated the progress which has been made toward more realistic load determinations.

It is appropriate to consider applying this accumulated wealth of research results to routine load problems. A modest attempt was made by the authors et al[1] previously. Steady-state load factors were modified to *include* transient effects, and special *allowances* were recommended for internal heat storages of the buildings. This procedure resulted in better accuracy than had been obtained by the steady-state method. But still, it could be considered only as an intermediate step. The thermal response method here discussed is a further development.

THERMAL RESPONSE METHOD

An electrical resistance-capacitance circuit is characterized by its response, in output current, to an input pulse which may be a temporary change of potential at the input side. It has been shown before[1] that building structures can be considered as resistance-capacitance circuits. A wall can be represented reasonably well by a π-circuit consisting of 3 resistances and 2 capacitances (Fig. 1). If constant temperature throughout the system is assumed, there can be no flow of heat. However, if the outdoor temperature is raised suddenly and after a short interval is reduced to the original level, heat will flow from the wall to the indoor air according to a definite sequence which is a characteristic of the wall circuit. This sequence is the thermal response of the wall to an outdoor temperature pulse or square wave. Fig. 2 shows a temperature pulse and a typical response curve.

Thermal response curves can be calculated readily as shown in Appendix A.

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[1] Exponent numerals refer to Bibliography.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956.

For general application, however, it is more convenient to use numbers than curves. A changing outdoor condition (*i.e.* temperature or solar radiation) may be considered as a sequence of square waves of constant duration and varying amplitude (Fig. 3). Accuracy of the calculations is best when the time intervals are very short, but is still adequate when they are each as long as 1 hr. This is convenient because weather data are recorded in hourly intervals by the U. S. Weather Bureau and are punched on business machine cards. These data can be used immediately as input information.

It is convenient to use the same length of periods for the response of the circuit as for the input pulses. The response of a structure (such as a wall or roof) on a unit basis, (*i.e.* per square foot area perpendicular to direction of heat flow and per

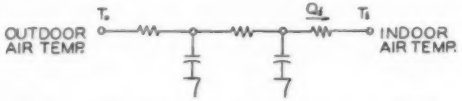

FIG. 1—THERMAL π-CIRCUIT OF A WALL

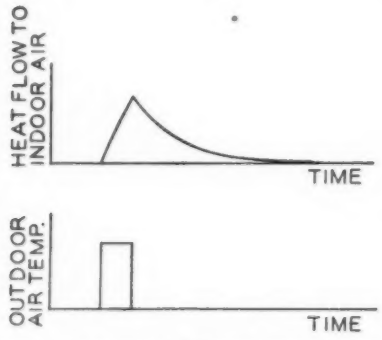

FIG. 2—OUTDOOR TEMPERATURE PULSE AND TYPICAL INDOOR HEAT FLOW RESPONSE

Fahrenheit degree difference between outdoor-indoor temperature) may now be expressed in response factors. Fig. 4 shows a typical sequence of response factors as result of a unit pulse. Furthermore, thermal systems, as considered here, are linear and the principle of super-position applies to them. Therefore, the resultant response to a sequence of input pulses is calculated by simply adding the hourly responses following each input pulse.

The heat flow through a structure can now be calculated for varying outdoor temperature and constant indoor temperature. The instantaneous value of heat flow per square foot for the hour n becomes:

$$q_n = (T_n - T_i)(F_1) + (T_{n-1} - T_i)(F_2) + (T_{n-2} - T_i)(F_3) + \ldots + (T_{n-m+1} - T_i)(F_x). \quad \ldots \quad (1)$$

and for the total structure:

$$Q_n = A(g_n) \quad \ldots \quad (2)$$

where

T = temperature.
F = response factor with subscripts denoting hour of response.
A = area of structure perpendicular to heat flow.
i = subscript denoting indoor condition.
n = subscript denoting hour for which calculation is made. n − 1 is preceding hour, and so on.
m = number of hours to have sufficient length of response for reasonable accuracy.

The discharge flow response to a unit input pulse is known as *transfer admittance* in electrical network theory. For heat transfer representation it may be called

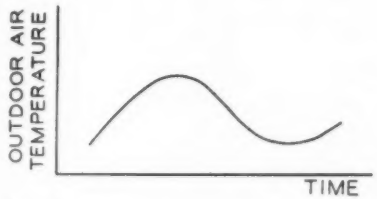

FIG. 3—TEMPERATURE CURVE AND EQUIVALENT SEQUENCE OF TEMPERATURE PULSES

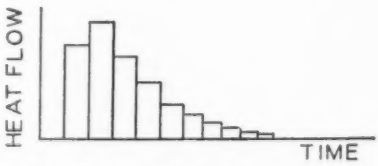

FIG. 4—SQUARE-WAVE REPRESENTATION OF HEAT FLOW RESPONSE FACTORS

transfer heat admittance. Accordingly, the response factors are *transfer heat admittance factors*. Since transfer heat admittance factors are a function of the structure only, they need to be calculated but once for each construction. They have permanent value and can be used from then on wherever the same construction is involved. Equation 1 is an example for calculation of transient heat flow by using the transfer heat admittance factors, F_1, F_2, and so on.

In many cases it is sufficient to calculate the heat flow into (or out of) a building for constant indoor temperature. Using transfer heat admittance factors for the respective structure, such calculations can be performed by hand, desk calculator or any digital computer according to the amount of calculations needed and the computing equipment available.

However, in other cases, it will be of interest to determine the change of indoor temperature as a function of heat flow. Again electrical network theory provides a practical way of solving this problem, *i.e.* by applying Thevenin's Theorem.[2] In terms of the heat load problem, Thevenin's Theorem can be explained as follows, referring to the wall circuit of Fig. 1.

If constant temperature throughout the system is again assumed there is no flow of heat. Heat flow can be initiated by changing either the outdoor or the indoor temperature. But, a temporary drop of indoor temperature will produce a different sequence of heat flow, from the interior surface of the wall to the indoor air, than would be caused by a rise of outdoor temperature. Heat flow response to an outdoor temperature pulse has been shown in Fig. 2. If the concept of heat flow response to a temperature pulse is applied to a drop of indoor temperature, the sequence will be as shown in Fig. 5. Positive values denote heat flow to the indoor air and negative values signify heat flow away from the indoor air. While the indoor temperature is down, heat flows out of the wall causing the temperature of the wall to drop. After the temperature pulse has terminated, and original indoor temperature is restored, the wall has temporarily a lower temperature than the indoor air. The wall must absorb heat from the indoor air to re-establish temperature balance.

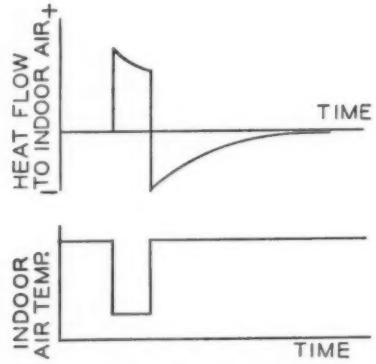

FIG. 5—INDOOR TEMPERATURE PULSE AND TYPICAL INDOOR HEAT FLOW RESPONSE

Both heat flow responses on unit temperature difference basis are admittances when viewed from the indoor air. Heat flow response to changing outdoor conditions, with indoor temperature kept constant, was previously called transfer heat admittance. In electrical network theory the term *driving point admittance* is used for the response to a unit change of potential at the observed point. For heat transfer problems it may be called *control point heat admittance*.

Thevenin's Theorem applied to the heat load problem says that the change of indoor air temperature with respect to a reference indoor temperature is a function of the 2 admittances. It is:

$$\Delta T_i = \Delta T_0 \, (\partial Q_i/\partial T_0)/\partial Q_i/\partial T_i \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In this equation, $\Delta T_0 \, (\partial Q_i/\partial T_0)$ is the total heat flow into (or out of) the building due to difference between outdoor and reference indoor temperature, ΔT_0, with $\partial Q_i/\partial T_0$ being the summation of the transfer heat admittance of all structures. The partial derivative $\partial Q_i/\partial T_i$ is the control point heat admittance of the entire building.

The transfer heat admittance $\partial Q_1/\partial T_0$ and control point heat admittance $\partial Q_1/\partial T_1$ are functions of the storage capacities of the structures. Considering a complete building it should be recognized that the control point heat admittance must include the branches for internal storage such as partitions and furniture. A change in indoor temperature will be followed by gain or release of heat by the internal storages. This is obviously not the case for the transfer heat admittance since by definition the indoor temperature is maintained constant. For calculation of control point heat admittance factors see Appendix A.

For machine calculations it is advisable to avoid divisions. Equation 3 therefore may be modified to read:

$$\Delta T_i = \Delta T_0 \, (\partial Q_i/\partial T_0)(\partial T_i/\partial Q_i) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The new term $\partial T_1/\partial Q_1$ is known in electrical network theory as *driving point impedance*. For thermal circuits it may be called *control point thermal impedance*. It would be difficult to calculate control point thermal impedance factors $\partial T_1/\partial Q_1$

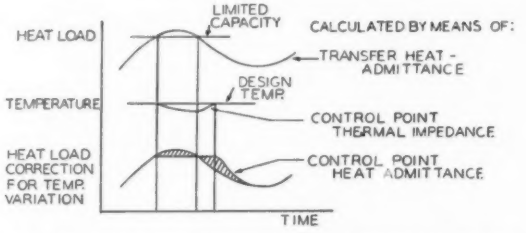

FIG. 6—SCHEMATIC DIAGRAM FOR CALCULATIONS BY THE THERMAL RESPONSE METHOD

from the differential equations for the entire building. However, the control point heat admittance factors $\partial Q_1/\partial T_1$ for the individual circuit branches can be added up to give the control point heat admittance for a complete building. From the control point heat admittance it is possible to calculate the control point thermal impedance by a process known as inverse convolution[3]. Details of the calculation are given in Appendix A.

Summarizing, calculations by the thermal response method may be performed in 3 distinct steps:

1. Calculation of heat load for varying outdoor conditions and constant indoor temperature.
2. Calculation of indoor temperature variation from design temperature due to changed control setting or as result of excess or insufficient capacity.
3. Calculation of heat load change as result of indoor temperature variations.

A schematic diagram of the 3 steps is given in Fig. 6.

The thermal response method can be used with precision. Only 2 factors influence the accuracy *viz.*: (*a*) the analogue representation of the thermal circuit and (*b*) the duration of each time element for input functions and response. Factor (*a*) may easily be checked by comparison of calculation with test results. As to the time element for input functions and response factors it is believed that the

selection of hourly values is a compromise satisfactory for most calculations with the exception perhaps of control system evaluation.

It may appear that the effort required to calculate thermal response factors for the many different constructions used throughout the country is too great. However, the response factors have permanent value and need to be calculated only once. Because this is so, it will be practical to use one of the large digital computers designed for engineering calculations. It would be only natural that tabulations of response factors would be published in appropriate handbooks such as THE GUIDE*.

EXPERIMENTAL PROOF OF THERMAL RESPONSE CALCULATIONS

As stated, there are 2 factors which may influence the accuracy of calculations by the thermal response method. It is, therefore, important to compare calculated data with test results. It is possible that laboratory methods for measuring tran-

TABLE 1—TRANSFER HEAT ADMITTANCE FACTORS FOR WALL OF RESEARCH RESIDENCE NO. 2, UNIVERSITY OF ILLINOIS

HOUR OF RESPONSE	BTU PER HR (F DEG) (1000 SQ FT)
n	5.99
n + 1	15.91
n + 2	12.47
n + 3	8.75
n + 4	6.07
n + 5	3.99
n + 6	3.04
n + 7	1.90
n + 8	1.33
n + 9	0.95

sient thermal response of structures will be developed. Measuring of thermal conductivities and film coefficients may be mentioned as precedents.

However, at present it is necessary to rely on comparisons of calculated values with heat flow measurements for which test conditions are sufficiently defined. This technique does not separate the effect of approximations in the analogue circuit from the effects of using finite rather than infinitesimal time intervals for the response factors.

In a previous study[1] test data of the University of Illinois, taken on their Research Residence No. 2[3], were compared with electrical analogue computer calculations which used a rather simple analogue circuit of the entire house. The calculated maximum load exceeded the corresponding test value by only 7 percent. This check was accepted as evidence that the analogue represented the house with good engineering accuracy.

In the present study test data for the west wall of the same Research Residence are used for comparison with calculated values. The wall was considered as π-circuit in the same manner as in the earlier study. Transfer heat admittance

* HEATING, VENTILATING, AIR CONDITIONING GUIDE (published annually by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, New York, N. Y.).

factors computed for the wall are listed in Table 1. The west wall was selected to avoid direct solar radiation effects. It was feared that the unavailability of local solar observations might limit the validity of the intended comparison. For the same reason the calculations were limited to the hours from 5:00 a.m. to 12 noon. Fig. 7 shows the result. The curve of test values was taken from Reference 3. It represents measurements by a heat flow meter installed in the indoor wall panels near their inside surfaces. The calculated values are for heat flow from the interior wall surface to the air. The calculated curves are much smoother than the test curves. During the very early morning hours of the test day, the outdoor temperature did not follow a smooth curve. These irregularities apparently pene-

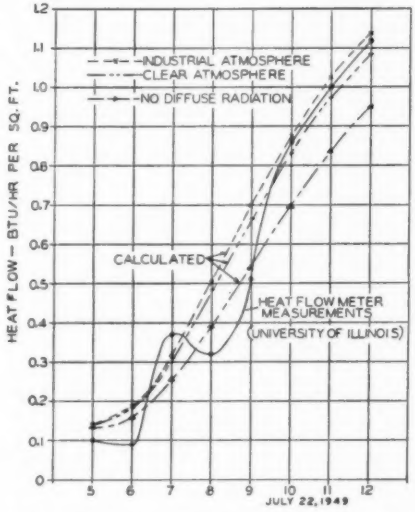

FIG. 7—COMPARISON OF CALCULATED RESULTS AND TEST DATA[3]

trated the wall up to the flow meter. The additional capacitance of the interior wall panel seems, however, to have been large enough to filter out that effect. Later, when the outdoor temperature followed a more regular course the flow meter data also became more uniform.

Without allowance for diffuse radiation, the calculated curve deviates from the measured more and more as the intensity of solar radiation rises during the day. Additional calculations were performed to include diffuse radiation. Since it was not known how clear the atmosphere was at the test location, data were used for both clear and industrial atmosphere respectively, as recorded in THE GUIDE[4]. It is felt that the results are gratifying and should create confidence in the thermal response method. It is hoped that future comparisons can be more extensive and made in greater detail.

CALCULATIONS BY THE THERMAL RESPONSE METHOD

Many different types of calculations can be based on the thermal response method. Determination of specific response factors for individual constructions, *i.e.* on unit area and unit temperature difference basis, is rather involved as shown in Appendix A. It is advisable to use one of the larger digital computers designed for engineering calculations. Unit response factors have permanent value since they can be used wherever the same constructions are involved. Calculations using unit response factors can be performed by hand, desk calculator or digital computers as available in accounting offices.

The first step in solving a transient heat flow problem is to reduce the thermal circuit of a building to its electrical analogue. The technique for doing this is the same as reported previously[1]. A single-story house with basement and attic was selected as an example. The analogue circuit was kept as simple as possible to

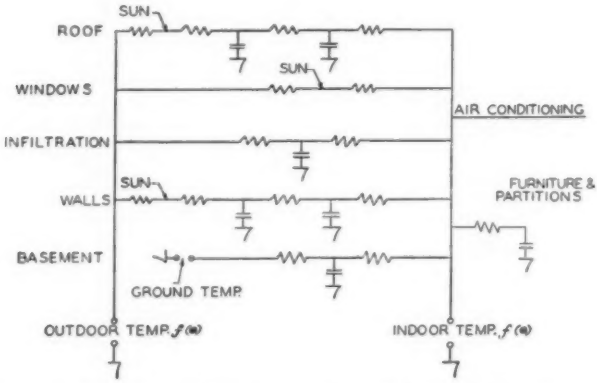

FIG. 8—SIMPLIFIED THERMAL CIRCUIT OF SINGLE-STORY HOUSE WITH BASEMENT

demonstrate the technique without complications of secondary importance. Based on the previous experience with analogue computer calculations, it is believed that the simplified circuit shown in Fig. 8 provides results sufficiently accurate for most purposes.

The sample house has 1800 sq ft of floor space. Walls are of frame construction with no insulation. They are represented by a π-circuit in Fig. 8. Windows and equivalent doors amount to 25 percent of the total wall area. It was assumed that Venetian blinds were fully drawn during the whole day, or that curtains or drapes would produce an equivalent thermal resistance when Venetian blinds were open. Accordingly, windows are shown as a double resistance circuit. Heat from solar radiation enters the circuit between the 2 resistances. The house has a pitched roof with natural ventilation of the attic space by louvers. The ceiling is insulated with 4-in. mineral wool blankets. Roof and ceiling are considered as 1 π-circuit. Infiltration (including forced ventilation) is represented by a T-circuit. A T-circuit, instead of a straight resistance, was considered appropriate since infiltration does not affect the temperature throughout the house instantly but rather goes

through a transient. The basement is assumed entirely below ground, and walls and floor are assumed exposed to a uniform ground temperature constant throughout the year. This is expressed in the diagram by a T-circuit with a constant potential at the ground terminal. Furniture and interior partitions form an L-circuit. This branch is a compromise lumping all interior storages (which actually have different thermal characteristics). The line labelled *air conditioning*, indicates an ideal year-round air-conditioning unit capable of maintaining a uniform and constant house temperature. Provisions for controls are, therefore, not shown in the circuit.

After the analogue circuit is laid out, the circuit parameters are determined. The individual thermal resistances and capacitances are calculated from the thermal conductivity and heat capacities of the individual structural materials used, and from the film coefficients for the various surfaces. Most of the resistances and

TABLE 2—TYPICAL RESPONSE FACTORS[a]

TRANSFER HEAT ADMITTANCE	BTU PER (HR) (F DEG)	HOUR n	n + 1	n + 2	n + 3	n + 4	n + 5	n + 6
Ceiling-Roof	Per 1000 Sq Ft	0.83	2.79	4.49	5.28	5.50	5.38	5.08
Wall	Per 1000 Sq Ft	31.40	67.10	59.29	38.10	23.37	14.21	8.63
Windows, Doors	Per Sq Ft	0.70	—	—	—	—	—	—
Infiltration	Per Air Change/Hr	99.8	114.6	42.2	15.5	5.7	2.1	0.8
Total House		409.2	202.8	123.8	72.2	44.6	29.4	20.6
CONTROL POINT HEAT ADMITTANCE								
Ceiling-Roof	Per 1000 Sq Ft	147	—10.4	—9.0	—7.7	—6.7	—5.8	—5.1
Wall	Per 1000 Sq Ft	827	—435	—77.4	—25.6	—13.1	—7.7	—4.7
Windows, Doors	Per Sq Ft	0.70	—	—	—	—	—	—
Infiltration	Per Air Change/Hr	280	—	—	—	—	—	—
Floor	Per 1000 Sq Ft	449.4	—38.94	—31.78	—25.94	—21.17	—17.28	—14.11
Partitions, Furniture	Per 1000 Sq Ft Floor Area	1196	—1018	—150.6	—22.22	—3.39	—	—
Total House		4800	—2462	—440	—133	—72	—56	—40
Control Point Thermal Impedance of Total House, per F deg per 1000 Btu per hr		0.2090	0.1115	0.0783	0.0575	0.0439	0.0348	0.0283

[a] Response factors listed in this table are for the house described in this paper.

capacitances in the circuit diagram are composite values for convenience of calculation. Transfer heat admittance factors and control point heat admittance factors are then computed for the individual circuit branches as outlined in Appendix A. As mentioned before, control point thermal impedance cannot be used for individual branches of a circuit. It is therefore necessary to calculate the control point heat admittance for the complete house and, using this value, to calculate the control point thermal impedance for the house. This procedure is also explained in Appendix A, and typical response factors are listed in Table 2.

A great advantage of the thermal response method is that actual weather data can be used as input functions. This makes the calculations realistic. Outdoor air temperature, solar radiation and wind velocity were included in the present calculations. The information was taken from business machine cards especially prepared by the U. S. Weather Records Center in Asheville, N. C. The card used is one entitled *Solar Radiation—Hourly Records* supplemented and extended with

information from a card entitled *WBAN Hourly Surface Observation*. Due to space limitations, this card is not shown here.

For thermal response calculations it is convenient to use solar radiation in the same manner as in the sol-air concept. This means calculating a fictitious temperature which would produce the same heat flow as solar radiation. It is

$$(\Delta T)_R = R_L K C_i \tau \epsilon (1/U) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where

$(\Delta T)_R$ = fictitious temperature elevation over free air temperature.
R_L = solar radiation in Langleys per hour.
K = conversion factor from Langleys per (hour) to Btu per (hour) (square foot).
C_i = incidence factor to transform horizontal radiation values to those for other surface angles.
τ = transmission factor (used for windows).
ϵ = absorptivity for solar radiation.
U = overall heat transfer coefficient between outdoor air and surface receiving solar radiation.

Wind conditions were considered only as they influence the rate of infiltration. A preliminary check indicated that the overall circuit response would not change appreciably with variations of outdoor film coefficients. Only for very low wind velocities, which occur rarely, could the accuracy be improved. However, the complication of variable film coefficients can hardly be justified. Accordingly, a wind velocity of 15 mph was assumed in selecting outside film coefficients. Infiltration, in contrast, is very sensitive to wind velocity. Expressed in terms of air changes per hour, infiltration can be assumed as a straight line function of wind velocity, above a minimum of ½ air changes per hour[6]. Response factors for infiltration differ from those applying to conduction loads. They are consequently expressed, on a unit basis, per degree outdoor-indoor temperature difference per air change per hour.

PEAK LOADS AND CUMULATIVE LOADS

One of the main objectives of load calculations is to determine the peak heating and cooling loads so that proper selection of equipment can be made. It is of interest also to obtain information on operating costs. These 2 objectives are met by a year-round heat load calculation for constant indoor temperature. Equations 1, 2 and 5 are used with transfer heat admittance factors for the respective constructions.

A year-round calculation was made for the house the analogue for which was drawn as Fig. 8. Weather records for New York City for the year from July 1, 1952 to June 30, 1953 were used. While that year may not have had extreme weather conditions, it serves for demonstration of the technique.

Calculations were performed on a 602A calculator in an accounting department. They were made in this sequence: conducted loads for walls and roofs, conducted plus radiation loads for windows and equivalent doors, infiltration load, and summary of all loads including allowance for basement load. Because of computing machine limitations, radiation on walls and roof was neglected and transients were limited to 6 hr. It should also be noted that the assumptions made for the basement result in a constant load for that branch when the indoor temperature is considered constant.

TABLE 3—PORTION OF CHART OF HOURLY HEATING LOADS TABULATED FROM BUSINESS MACHINE CARDS

LOCATION	CONSTRUCTION	YEAR	MONTH	DAY	HOUR	OUTDOOR TEMP.	RADIATION ON HORIZ. SURFACE	WIND (MPH)	INDOOR TEMP.	WALL AND ROOF LOADS	COMPUTING FACTOR	WINDOW LOAD	COMPUTING FACTOR	INFILTRATION LOAD	TOTAL LOAD	MARKER	DAILY TOTAL HEATING LOAD	DAILY TOTAL COOLING LOAD
1	14B1A	52	12	27	16	33	1.5	20	75	13128—		11298—	65—	11145—	43701—			
1	14B1A	52	12	27	17	30		35	75	13575—		12105—	137—	16503—	50313—	4		
1	14B1A	52	12	27	18	28		36	75	14166—		12043—	148—	22954—	57893—	5		
1	14B1A	52	12	27	19	25		30	75	14897—		13450—	128—	24493—	60975—	6		
1	14B1A	52	12	27	20	22		32	75	15709—		14257—	145—	25284—	63380—	6		
1	14B1A	52	12	27	21	20		30	75	16537—		14795—	140—	26088—	65570—	6		
1	14B1A	52	12	27	22	19		30	75	17349—		15064—	143—	26181—	66724—	6		
1	14B1A	52	12	27	23	17		30	75	18023—		15602—	148—	26676—	88431—	7	1,046,901—	
1	14B1A	52	12	28		17		35	75	18593—		15602—	177—	29095—	71420—	7		
1	14B1A	52	12	28	1	17		30	75	18973—		15602—	148—	29502—	72207—	7		
1	14B1A	52	12	28	2	16		38	75	19220—		15371—	198—	31522—	74743—	7		
1	14B1A	52	12	28	3	15		30	75	19453—		16140—	153—	31805—	75528—	7	←PEAK HEATING LOAD	
1	14B1A	52	12	28	4	14		29	75	19717—		16409—	149—	29343—	73599—	7		
1	14B1A	52	12	28	5	14		27	75	19944—		16409—	137—	27359—	71842—	7		
1	14B1A	52	12	28	6	13	.4	26	75	20162—		16678—	133—	25817—	70737—	7		
1	14B1A	52	12	28	7	13		21	75	20373—		16678—	102—	22880—	58081—	7		
1	14B1A	52	12	28	8	13	9.6	19	75	20521—	35	7263—	90—	19472—	55386—	5		
1	14B1A	52	12	28	9	15	22.7	20	75	20525—	53	1883—	93—	17805—	48343—	4		

A year-round constant indoor temperature of 75 F was selected for the example. Important calculation results were printed on strip charts. Table 3 is a chart section for the peak heating load extended to include several hours before and after. More extended charts for both the heat peak load and the cooling peak load are included in the Appendix as Tables C-1 and C-2. For convenience in finding high load conditions special markers were used in these charts (see Table 3) with the following meanings:

LOAD	PERCENT OF PEAK LOAD
1: sensible cooling load above	90
2: sensible cooling load above	80
3: sensible cooling load above	70
4: heating load above	60
5: heating load above	70
6 heating load above	80
7: heating load above	90

Neither peak heating nor cooling load coincided with the extreme temperatures of the year. The maximum sensible cooling load (Table C-1) occurred on September 13, 1952 at 3:00 p.m.; at which time the outdoor temperature was 93 F as compared to extremes of 96 F on several days in July. However, the solar altitude on September 13 is lower than in July, producing a considerably higher radiation load on the house and accounting for the peak load.

Maximum heating load occurred on December 28, 1952 at 3:00 a.m. (Table 3). Again the outdoor temperature of 15 F was less severe than the extreme of 13 F several hours later. But at the peak load conditions a wind velocity of 30 mph raised the infiltrated load sufficiently (31800 Btu/hr as against 25,800 Btu/hr) to raise the heating load to a maximum.

Having found the extreme loads, a comparative calculation was made by hand for these hours. While transfer heat admittance factors for 6 hr were used in the computer calculations, they were extended to 7 hr for the hand calculations. Furthermore, solar radiation effects were included in the calculations of wall and roof loads. A comparison of hand calculations and computer results is given in Table 4. The error due to neglecting solar radiation on roof and wall is 15.4 percent. This is not consistent with the precision obtainable with thermal response calculations. Therefore, solar radiation on wall and roof ought to be included in future machine computations. Comparison of 7-hr and 6-hr transients shows a difference of 11.9 percent for the ceiling-roof circuit. This is not surprising since the response at the 6th hour is barely beyond its peak due to the high time constant of that circuit. But, due to the low absolute value of this branch, the effect on the total result is only 0.3 percent.

Hourly heating and cooling loads were added up to provide daily, monthly and yearly cumulative values. Monthly and yearly totals are listed in Table 5. Part load requirements can also be evaluated easily from year-round load data. Figs. 9 and 10 list as ordinates the number of hours for which the load is equal to or in excess of the percentage of peak load used as abscissae. Cumulative values of part loads are shown in Figs. 11 and 12 as bar charts. Increments of 10 percent each are used as abscissae. Since the computer calculated only sensible loads, they were adjusted to include the latent portion. A total of 12,000 Btu/hr was assumed for a sensible cooling load of 8600 Btu/hr.

TABLE 4—PEAK LOADS—BREAKDOWN BY CIRCUIT BRANCH AND CALCULATION DETAILS

TIME OF PEAK LOAD	CIRCUIT BRANCH	BTU PER HR		EFFECT OF RADIATION			DIFFERENCE OF 7 HR vs. 6 HR TRANSIENT CONSIDERATION		
		COMPUTER	HAND CALCULATED	BTU/HR	PERCENT OF BRANCH LOAD	PERCENT OF TOTAL LOAD[a]	BTU/HR SENS.	PERCENT OF BRANCH LOAD	PERCENT OF TOTAL LOAD[a]
	Walls	5,081	6,451 = 8,566	1,853	28.7	6.2	96	2.1	0.4
	Roof		2,115	1,426	68.4	4.8	82	11.9	0.3
9-13-52 3:00 p.m.	Windows	17,485	17,542	12,700	72.4	42.7	—	—	—
	Floor	— 8,130	— 8,130	—	—	—	—	—	—
	Infiltration	3,325	3,333	—	—	—	13[b]	0.4[b]	<0.1[b]
	Total, sens.	17,761	21,311	15,979	—	53.7	191	—	0.9
	Total[a]	24,782	29,735	—	—	—	—	—	—
	Walls	−19,453	−17,499	—	—	—	589	3.4	0.8
	Roof		— 3,024	—	—	—	501	16.6	0.7
12-28-52 3:00 a.m.	Windows	−16,140	−16,140	—	—	—	—	—	—
	Floor	— 8,130	— 8,130	—	—	—	—	—	—
	Infiltration	−31,805	−32,036	—	—	—	207[b]	0.6[b]	0.3[b]
	Total	−75,528	−76,829	—	—	—	1,297	—	1.7

[a] Sens + latent.
[b] Difference of 7 hr *vs* 5 hr transient consideration.

TABLE 5—MONTHLY AND YEARLY TOTAL HEATING AND COOLING LOADS

MONTH	HEAT LOAD BTU	SENSIBLE COOLING LOAD, BTU	SENSIBLE PLUS LATENT LOAD BTU[a]
July '52	2,972,860	1,433,521	2,000,264
Aug.	4,948,572	694,127	968,550
Sept.	6,785,790	709,576	990,107
Oct.	14,514,611	99,021	138,169
Nov.	18,452,044	8,553	11,934
Dec.	26,003,740	—	—
Jan. '53	26,125,341	—	—
Feb.	22,555,956	—	—
Mar.	22,023,762	—	—
Apr.	17,397,676	6,088	8,495
May	11,897,975	111,170	155,121
June	5,679,422	839,763	1,171,763
Total year	179,357,749	3,901,819	5,444,403

[a] Allowance for latent load is made by rating total load at 12,000 Btu per hr for 8600 Btu per hr calculated sensible load.

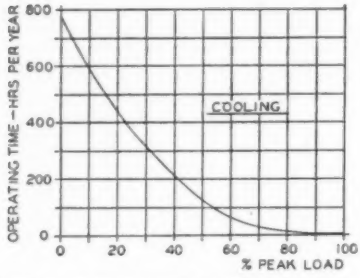

FIG. 9—CUMULATIVE HOURS OF COOLING OPERATION AT OR AT MORE THAN STATED LOAD

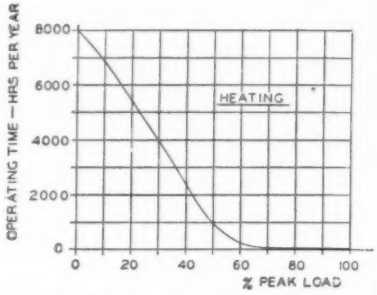

FIG. 10—TOTAL COOLING DEMAND IN 10 PERCENT INCREMENTS OF PEAK LOAD

VARIATION OF INDOOR AIR TEMPERATURE

In the preceding calculations it was assumed that the indoor air temperature was kept constant by heating and cooling equipment which matched the load demand perfectly at all times. In many cases it will be of interest to know how the indoor temperature would deviate from the design temperature when the capacity is less than, or more than, design value, and how temperature and load would follow a changed control setting.

Temperature variations are calculated by using control point thermal impedance factors for the entire house. The difference between heat load and actual available capacity can be treated like a transfer heat admittance. Accordingly, using this difference for the term $\Delta T_0 \ (\partial Q_1/\partial T_0)$, Equation 4 gives the indoor temperature deviation due to insufficient or excess capacity.

A change of temperature is followed in turn by a change of heat flow. Corrections for heat flow values are calculated using control point heat admittance factors. These are identical with the basic heat load calculations, Equations 1 and 2.

Temperature variations due to insufficient capacity were calculated by hand for the house discussed for the night of peak heating load. Fig. 13 shows the indoor

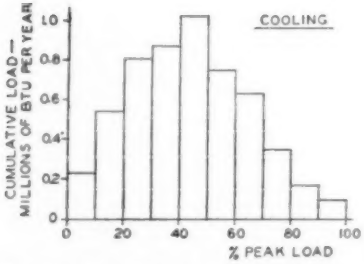

FIG. 11—CUMULATIVE HOURS OF HEATING OPERATION AT OR AT MORE THAN STATED LOAD

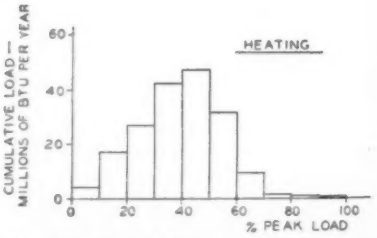

FIG. 12—TOTAL HEATING DEMAND IN 10 PERCENT INCREMENTS OF PEAK LOAD

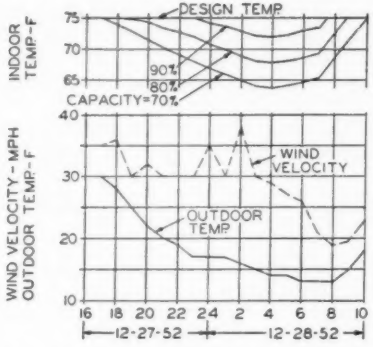

FIG. 13—VARIATION FROM DESIGN INDOOR TEMPERATURE FOR VARIOUS PERCENT DESIGN CAPACITY. CORRESPONDING OUTDOOR TEMPERATURES AND WIND VELOCITIES ARE SHOWN IN LOWER PORTION OF GRAPH

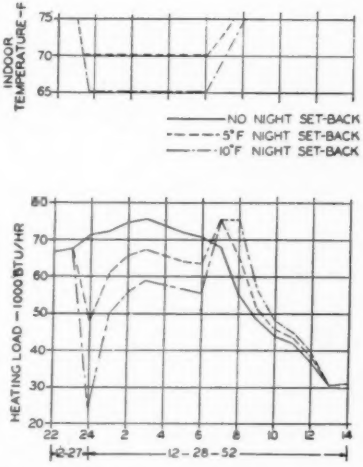

FIG. 14—INDOOR TEMPERATURE AND HEAT LOAD AS FUNCTIONS OF TIME AND NIGHT SETBACK, ASSUMING IDEAL HEATING EQUIPMENT CONTROLLED BY IDEAL THERMOSTAT

temperature as a function of time for 3 different percentages of capacity available *viz.*: 90, 80 and 70 percent of peak load. Outdoor temperature and wind velocity are also plotted. It can be seen that lowest indoor temperature did not occur at the time of lowest outdoor temperature, but was influenced more strongly by very high wind velocity.

Fig. 14 demonstrates the effect of thermostat setback on temperature and heat load for the same night as considered for Fig. 13. Cumulative loads corresponding

TABLE 6—HEAT LOAD APPLICATION EQUATIONS. LOADS IN BTU PER HOUR PER SQUARE FOOT

Region: New York, N. Y.

CONSTRUCTION	LOAD	ORIENTATION	NO INSULATION	2 IN. MIN WOOL INS.	4 IN. MIN WOOL INS.
Frame Wall: Wood siding; Fir sheathing plus building paper $^{20}/_{60}$ in.; 2 x 4 in. studding; Gypsum lath $^{3}/_{8}$ in.; plastered, $^{1}/_{2}$ in.	Cooling	North East South West	21.86–0.242 T_i 22.20–0.242 T_i 25.10–0.242 T_i 24.19–0.242 T_i		
	Heating	—	4.04–0.242 T_i		
Pitched Roof — Ceiling[a]: Asphalt shingles; fir sheathing plus building paper $^{20}/_{60}$ in.; 2 x 6 in. rafters; Attic air space naturally ventilated; 2 x 6 in. joists; Gypsum lath $^{3}/_{8}$ in.; plastered, $^{1}/_{2}$ in.	Cooling	—			3.37–0.0293 T_i
	Heating	—			0.52–0.0293 T_i
Floor over Basement[a] (entirely below ground): Hardwood $^{13}/_{16}$ in.; building paper; subflooring, $^{20}/_{60}$ in.; 2 x 8 in. joists; Basement wall 12 in. cement blocks; Floor 3 in. concrete.	Cooling	—	13.31–0.238 T_i		
	Heating	—	13.31–0.238 T_i		
			No shades	Venetian blinds drawn	Awnings
Windows: Single pane	Cooling	North East South West		65.15–0.701 T_i 65.15–0.701 T_i 115.6 –0.701 T_i 170.9 –0.701 T_i	
	Heating	—		10.51–0.701 T_i	
			$^{1}/_{2}$ Air Change/hr[b]	$^{3}/_{4}$ Air Change/hr[b]	1 Air Change/hr[b]
Infiltration[a] Windows = 25 percent of wall area	Cooling Heating	— —	9.94–0.108 T_i 4.81–0.302 T_i		

[a] Load in Btu/hr per sq ft of floor area.
[b] Minimum rate of air changes.

to constant temperature and to night setback may be compared to indicate that savings of approximately 5 percent for 5 F and 11 percent for 10 F setback can be realized under the conditions of the example.

APPLICATION METHOD

One purpose of application calculations is to determine the maximum load conditions of a building. The selection of heating and cooling equipment is based on application calculations. Customer satisfaction depends largely on proper sizing of the equipment. It is believed that the user can be served best if the load cal-

culations are exact and the possibility of errors is kept to a minimum. Load calculations which are based on actual weather data (*i.e.* temperature, solar radiation and wind conditions), and which account properly for the thermal transient characteristics of the structures, offer the greatest opportunity for success. If the calculations are reasonably simple the margin of error will also be reasonable.

The thermal response method makes possible a uniquely superior application method. The calculations are based on actual weather data, and transient characteristics of structures are included in the response factors. It seems to be logical to develop regional tables of heat load factors for the commonly used constructions. These factors incorporate both weather conditions and the transient characteristics

TABLE 7—FORM SHEET FOR HEAT LOAD CALCULATIONS

Location: New York, N. Y. *Indoor Temperature: 75 F*

			AREA, SQ FT			FACTOR COOLING	COOLING LOAD	FACTOR HEATING	HEATING LOAD
Walls	N	Gross Wall	482	—	—	—	—	—	—
		Window	—	176	—	12.57	2200	—	—
		Net Wall	—	—	306	3.71	1130	—	—
	E	Gross Wall	329	—	—	—	—	—	—
		Window	—	12	—	12.57	150	—	—
		Net Wall	—	—	317	4.05	1280	—	—
	S	Gross Wall	485	—	—	—	—	—	—
		Window	—	146	—	63.00	9200	—	—
		Net Wall	—	—	339	6.95	2350	—	—
	W	Gross Wall	328	—	—	—	—	—	—
		Window	—	50	—	118.34	5900	—	—
		Net Wall	—	—	278	6.04	1680	—	—
	Total	Window	—	384	—	—	—	—42.07	—16200
		Net Wall	—	—	1240	—	—	—14.11	—17500
Roof		Floor	1800	—	—	1.17	2100	—1.68	—3020
Floor		Floor	1800	—	—	—4.54	—8150	—4.54	—8150
Infiltration		Floor	1800	—	—	1.84	3310	—17.84	—32100
						Total sens Cooling: 21150		Total Heating:	—77170

of the structure. Expressions similar to equations 1 and 2 can be reduced, for application purposes, to the form:

$$Q = (a - bT_i)\,A \quad . \quad (6)$$

where

Q = heat load of structure (such as individual walls, roof).
A = area of structure perpendicular to heat flow.
T_i = design indoor temperature.
a and b = Factors typical for structure and weather conditions of the region.

Values for the parenthetical portion of Equation 6, called *application equations* for short, have been calculated for the construction details of the house used as an example throughout this study. The factors incorporate the weather conditions which produced the peak loads of the sample calculations. For future application equations it would be advisable to study weather data of a number of years to find the true extremes. Table 6 is a sample table of application equations which could be published in a handbook.

Another possibility would be to solve the equations for typical design indoor temperatures and to list regional load factors in tables of handbooks. Table B-1 in Appendix B is an example.

In the field it would be desirable to have a simple form sheet. Table 7 is a sample of a form sheet for load calculations by the thermal response method. The example included there is again the sample house. Areas are multiplied by the corresponding load factors which would be taken from sources like Table 6 or Table B-1. Columns are then added vertically to give total loads.

CONCLUSIONS

The *thermal response method* described here has a number of unique features which make it appear superior to other methods of heat load calculation:

1. A high degree of accuracy is reached by the use of actual weather data (temperature, solar radiation and wind conditions) as recorded by the U. S. Weather Bureau. Selection or assumption of outdoor conditions is avoided.
2. A high degree of accuracy is made possible by utilizing experience from electrical analogue studies and by representing thermal transients of structures by sequences of response factors.
3. The method is practical for engineering and application purposes because calculations using response factors can be made by hand, by desk calculators or by digital computers as available in accounting departments. Response factors have permanent value and may be made available through handbooks.
4. Transient temperature variations of indoor air as a result of capacity changes or control setting can also be calculated by using appropriate response factors.
5. A simple application method can be developed which will avoid the customary dependence on outdoor *design temperatures* and other experience factors.

ACKNOWLEDGMENT

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APPENDIX A

Response Factors for Thermal Response Method

The Unit Thermal Response: The unit thermal response is a sequence of numbers which define the transient response of a thermal circuit to a unit input pulse. The numbers are the values of the response at successive time intervals. In the thermal circuits of engineering interest this sequence of numbers decays to negligible value after a reasonable length of time and the sequence may be terminated. This Appendix describes the methods which have been used to calculate unit thermal responses.

As a practical example the wall circuit of Fig. A-1 will be used. In this circuit the R's and C's represent thermal resistances and heat capacities. The equation of heat balance applies at both of the junctions a and b. The differential equations which result from applying the condition of heat balance are

$$\frac{T_0 - T_a}{R_1} + \frac{T_b - T_a}{R_2} = C_1 \frac{d\,T_a}{d\,\theta} = C_1\, p\, T_a \quad \text{(A-1)}$$

and

$$\frac{T_a - T_b}{R_2} + \frac{T_i - T_b}{R_3} = C_2 \frac{d\,T_b}{d\,\theta} = C_2\, p\, T_b \quad \text{(A-2)}$$

where

$$p = \frac{d}{d\,\theta} \text{ differential operator}$$

$$T_0,\ T_a,\ T_b,\ T_i = \text{temperatures}$$

$$\theta = \text{time}$$

These 2 differential equations contain all of the information necessary to calculate the transfer heat admittance and the control point heat admittance.

The Transfer Heat Admittance: The transfer heat admittance is defined as the heat flow at the control point (indoor air temperature bus) which results from a unit change in the outdoor temperature, the indoor air temperature being held constant at reference temperature. For convenience the following symbols are used

$$Q_i = \frac{T_b - T_i}{R_3} = \text{heat flow to control point, Btu per hr} \quad \text{(A-3)}$$

$$\Delta T_0 = T_0 - T_i. \quad \text{(A-4)}$$

T_i is considered constant

Then,

$$\text{Transfer Heat Admittance} = \frac{Q_i}{\Delta T_0} \quad \text{(A-5)}$$

Combining Equations A-1, A-2 and A-5,

$$\frac{Q_i}{\Delta T_0} = \left(\frac{1}{R_1 + R_2 + R_3}\right)\left(\frac{1}{1 + p\alpha + p^2\beta}\right)$$

$$= \left(\frac{1}{R_1 + R_2 + R_3}\right)\left(\frac{1}{(1 + p\tau_1)\,(1 + p\tau_2)}\right) \quad \text{(A-6)}$$

where

$$\alpha = \frac{C_1 R_1 (R_2 + R_3) + C_2 R_3 (R_1 + R_2)}{R_1 + R_2 + R_3} \quad \text{(A-7)}$$

$$\beta = \frac{C_1 C_2 R_1 R_2 R_3}{R_1 + R_2 + R_3} \quad \text{(A-8)}$$

$$\tau_1, \tau_2 = \text{roots of } 1 + p\alpha + p^2\beta \quad \text{(A-9)}$$

The roots τ_1 and τ_2 must be real[2]. They are the thermal time-constants of the circuit under consideration, and have the same analogous significance as electrical or mechanical time-constants. The factor $\left(\frac{1}{R_1 + R_2 + R_3}\right)$ will be recognized as the steady state thermal conductance of the circuit. The time-constants are numbers characteristic of the nature of the transient response of the network. In general there will be as many time-constants as there are storage elements in thermal circuits of this type.

Equation A-6 may be rewritten in the equivalent form

$$\frac{Q_1}{\Delta T_0} = \left(\frac{1}{R_1 + R_2 + R_3}\right)\left(\frac{A}{1 + p\tau_1} + \frac{B}{1 + p\tau_2}\right) \quad \text{(A-10)}$$

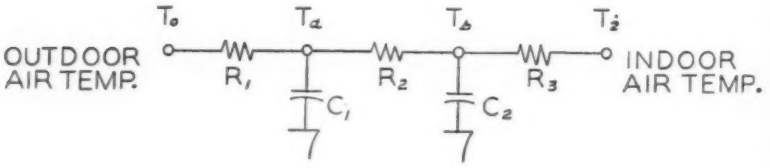

FIG. A-1—WALL CIRCUIT USED AS EXAMPLE

where

$$A = -\frac{\tau_1}{\tau_2 - \tau_1} \quad \text{(A-11)}$$

$$B = \frac{\tau_2}{\tau_2 - \tau_1} \quad \text{(A-12)}$$

This form of the equation is used to simplify the actual calculation of the response by clearly separating the response into the sum of 2 components, one associated with τ_1 and the other with τ_2.

The solution of differential equations of the form of Equation A-10 is very well known and discussed in the majority of textbooks on engineering mathematics or circuit theory[6, 7]. For this reason a detailed discussion of the solution is omitted. For the purposes of explaining this overall calculation it can be stated that each of the components of the form $\frac{1}{1 + p\tau}$ has a transient solution of the form shown in Fig. A-2. The transient builds up according to the equation

$$\text{Output} = (1 - e^{-\theta/\tau}) \quad \text{(A-13)}$$

for the duration of the input impulse (till $\theta = 1$) and then decays exponentially by the factor $e^{-(\theta-1)/\tau}$. This latter decay factor has the exponent, $-\frac{\theta - 1}{\tau}$, because it is initiated one unit of θ later in time than the initiation of the buildup.

This expression may appear formidable at first glance. This is an illusion caused by the notation which must be used to describe the effect. Note that the transient builds up according to Equation A-13 till $\theta = 1$. The quantity in parenthesis then remains constant at the value attained at $\theta = 1$. From that time on the quantity decays exponentially, so that the ratio of the $(n + 1)$th value to the nth value is given by

$$\frac{(1 - e^{-1/\tau})\, e^{-[(n+1)-1]/\tau}}{(1 - e^{-1/\tau})\, e^{-(n-1)/\tau}} = e^{-1/\tau} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-14)}$$

where n = number of time interval $(n > 1)$
This means that all values of the transient can be evaluated at the time instants marked by X's in terms of the numerical value of $e^{-1/\tau}$.

As an example of this, assume that the time unit is one hour and that the thermal time-constant is one hour. Then the value of $e^{-1/\tau}$ is

$$e^{-1/\tau} = e^{-1} = 0.368 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-15)}$$

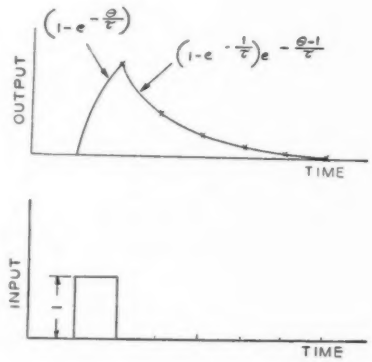

FIG. A-2—OUTDOOR TEMPERATURE PULSE AND TYPICAL INDOOR HEAT FLOW RESPONSE

From Equation A-13 the value of the response at the first hour interval will be

$$(1 - e^{-1/\tau}) = (1 - 0.368) = 0.632 \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-16)}$$

From Equation A-14 the value of the response at the second hour interval will be

$$(0.632)\, e^{-1/\tau} = 0.632 \times 0.368 = 0.232 \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-17)}$$

and the third hour interval

$$(0.232)\, e^{-1/\tau} = 0.232 \times 0.368 = 0.0855. \quad . \quad . \quad . \quad . \quad . \quad \text{(A-18)}$$

and so on, giving the response shown in Fig. A-3.

At this time it is important to reconsider the nature of the thermal response method. When the method was described it was assumed that the input value could be multiplied by a number which would give a correct value of response for the time interval considered. It is apparent from the response curve of Fig. A-3 that the instantaneous

heat flow can change considerably during one time interval. Therefore, the number to be used must be a mean value. The area under the heat flow response curve is the total amount of heat transferred, while the area under the curve between adjacent time intervals is the amount of heat transferred during the interval considered. It is apparent that the value used for the response should be the area under the true transient response curve for the corresponding time interval.

The areas involved are readily obtained from the values of the response curve previously calculated (Equations A-16, A-17 and A-18 in this example). The area under a curve is the integral of the function defining the curve over the desired interval. For the transient response considered here, the area under the curve for the first interval will be

$$\text{Area} = \int (1 - e^{-\theta/\tau})\, d\theta = \theta \Big|_0^1 + \tau e^{-\theta/\tau} \Big|_0^1 = 1 - \tau(1 - e^{-1/\tau}) \quad . \quad . \quad \text{(A-19)}$$

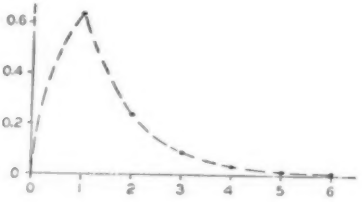

FIG. A-3—RESPONSE CURVE CALCULATED FOR TIME UNIT OF ONE HOUR AND THERMAL TIME-CONSTANT OF ONE HOUR

It should be noticed that the quantity in parenthesis is the previously calculated value from Equation A-16.

In a similar fashion the areas under the succeeding time intervals may be obtained, since

$$\text{Area} = \int_{\theta_1}^{\theta_2} K e^{-\theta/\tau}\, d\theta = -K\tau e^{-\theta/\tau} \Big|_{\theta_1}^{\theta_2} = K\tau(e^{-\theta_1/\tau} - e^{-\theta_2/\tau}) \, . \quad \text{(A-20)}$$

where,

$$K e^{-\theta_2/\tau} = \text{value of function at } \theta_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-21)}$$

$$K e^{-\theta_1/\tau} = \text{value of function at } \theta_1 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-22)}$$

Hence:

$$\text{Area} = (\tau)\ (\text{difference in value at the successive intervals}) \quad . \quad . \quad \text{(A-23)}$$

Equations A-19 and A-20 are the equations used to obtain the values of the unit response which in turn is used for calculating the transient heat flow resulting from changes in outdoor air temperature. Fig. A-4 shows the values obtained for the same example as that of Fig. A-3. The values are shown as lines since they are assumed to hold for the interval under consideration.

These calculations are typical of one form of manual calculation which has been used to obtain unit response factors. The same result could have been obtained by other calculations, and there is no assurance that the method described is most efficient or even particularly suitable for use with automatic data processing machinery.

The response factors of the type shown in Fig. A-4 are multiplied by the constants $\frac{A}{R_1 + R_2 + R_3}$ or $\frac{B}{R_1 + R_2 + R_3}$ to form the transfer heat admittance factor indicated by Equation A-10.

The Control Point Heat Admittance: The control point heat admittance is defined as the heat flow at the control point which results from a unit change in control point temperature, the outdoor air temperature being held constant at reference temperature. If the symbol ΔT_i is used to designate the change in control point temperature with T_0 considered constant, then

$$\text{Control Point Heat Admittance} = \frac{Q_1}{\Delta T_i} \quad . \quad . \quad . \quad . \quad \text{(A-24)}$$

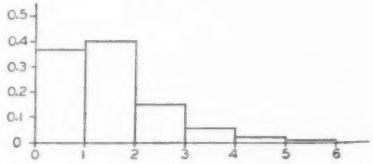

FIG. A-4—VALUES OF UNIT RESPONSE FOR SAME CONDITIONS USED IN FIG. A-3

Combining Equations A-1, A-2 and A-24 we obtain

$$\frac{Q_i}{\Delta T_i} = -\frac{1}{R_1 + R_2 + R_3}\left(\frac{1 + p\gamma + p^2\delta}{1 + p\alpha + p^2\beta}\right)$$

$$= -\left(\frac{1}{R_1 + R_2 + R_3}\right)\left(\frac{1 + p\gamma + p^2\delta}{(1 + p\tau_1)(1 + p\tau_2)}\right) \quad . \quad . \quad . \quad . \quad \text{(A-25)}$$

where,

α, β, τ_1 and τ_2 have their earlier definitions

and

$$\gamma = C_1 R_1 + C_2 (R_1 + R_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-26)}$$

$$\delta = C_1 C_2 R_1 R_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-27)}$$

Equation A-25 may be arranged to give the equivalent form

$$\frac{Q_i}{\Delta T_i} = -\frac{1}{R_1 + R_2 + R_3}\left[A + \frac{B}{1 + p\tau_1} + \frac{C}{1 + p\tau_2}\right] \quad . \quad . \quad \text{(A-28)}$$

where

$$A = \frac{\delta}{\tau_1 \tau_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(A-29)}$$

$$B = \frac{\tau_1(\tau_1 - \gamma) + \delta}{\tau_1(\tau_1 - \tau_2)} \quad \text{(A-30)}$$

$$C = \frac{\tau_2(\gamma - \tau_2) - \delta}{\tau_2(\tau_1 - \tau_2)} \quad \text{(A-31)}$$

Equation A-28 is quite similar to Equation A-10 and will generally be handled by the same methods. The first term represents a transient of the same form as the input signal, a unit pulse in the case of interest. A typical transient is shown in Fig. A-5.

The effect of the pulse component is readily seen. The significance of negative flow values is to represent heat flow away from the control point. While differing sharply in detail from the factors considered previously, the same comments and techniques apply.

The Control Point Thermal Impedance: In electrical network theory it is convenient to use quantities which are the reciprocals of admittances. These quantities are generally

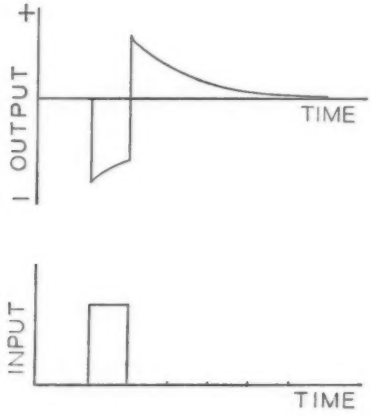

FIG. A-5—INDOOR TEMPERATURE PULSE AND TYPICAL INDOOR HEAT FLOW RESPONSE

called impedances. Resistance is the reciprocal of conductance and is a special form of impedance. As long as the problem of immediate concern is the heat flow resulting from temperature (or sol-air temperature) changes, the admittance concept is adequate and actually convenient, since the admittance of a number of parallel paths is the sum of the individual admittances. However, the inadequacy of the admittance concept becomes apparent when it is desired to calculate the temperature changes at the control point as a result of transient heat unbalance; for example, the operation of a simple on-off heating or cooling unit. This calculation can be made by the same process as was used to determine the heat flow provided the proper unit thermal response factors are used,—in this case the control point thermal impedance.

The control point thermal impedance is defined as the temperature change at the control point which results from a unit change in heat flow to the control point, the heat flow being considered at reference temperature. It should be noted that the control point thermal impedance is a characteristic of the total thermal circuit which *is not* the sum of the effects of the separate parallel branches of the circuit.

The problem of calculating the control point thermal impedance directly from the thermal circuit constants becomes formidable except in the simplest of circuits. For this reason an indirect approach was taken and has proved convenient.

Action and reaction are reversed in such a way that the values of temperature and heat flow are not affected. It was stated before that the control point heat admittance of the entire structure is the sum of the admittances of all parallel branch circuits. The temperature response at the control point to a unit pulse of heat flow at the control point must be the same as the temperature change which would produce the unit pulse of heat flow. Using the symbol $\frac{T_i}{\Delta Q_i}$ to denote the control point thermal impedance the reciprocity can be expressed as:

$$\left(\frac{T_i}{\Delta Q_i}\right) \times \left(\frac{Q_i}{\Delta T_i}\right) = 1, 0, 0, 0, 0 \ldots \ldots \ldots \quad \text{(A-32)}$$

where 1, 0, 0, 0, 0 . . . . . is the number sequence which is equivalent to the unit pulse. Therefore,

$$\frac{T_i}{\Delta Q_i} = \frac{1, 0, 0, 0, 0 \ldots}{\frac{Q_i}{\Delta T_i}} \ldots \ldots \ldots \quad \text{(A-33)}$$

The division indicated is performed in the sense of algebraic division.

APPENDIX B

Tabulation of Handbook Data

Table B-1 shows an alternate method of tabulating Heat Load Factors to the method illustrated by Table 6 of the paper.

TABLE B-1—HEAT LOAD FACTORS FOR WALLS, BTU PER HR (SQ FT)
Region: New York, N. Y.

CONSTRUCTION	LOAD	ORIENTATION	NO INSULATION			2 IN. MIN WOOL INS.			4 IN. MIN WOOL INS.		
			INDOOR TEMPERATURE			INDOOR TEMP.			INDOOR TEMP.		
			70	75	80	70	75	80	70	75	80
Frame Wall: Wood siding; Fir sheathing plus building paper $\frac{25}{64}$ in.; 2 x 4 in. Studding Gypsum lath, $\frac{3}{8}$ in.; Plastered, $\frac{1}{2}$ in.	Cooling	North	4.92	3.71	2.50						
		East	5.26	4.05	2.84						
		South	8.16	6.95	5.74						
		West	7.25	6.04	4.83						
	Heating	—	−12.90	−14.11	−15.32						
Brick Veneer: Brick, 4 in.; Fir sheathing plus building paper $\frac{25}{64}$ in.; 2 x 4 in. Studding; Gypsum lath, $\frac{3}{8}$ in.; Plastered $\frac{1}{2}$ in.	Cooling	North									
		East									
		South									
		West									
	Heating	—									
Masonry: Brick, 8 in.; Air Space; 1 in. Furring Strip; Gypsum lath $\frac{3}{8}$ in.; Plastered, $\frac{1}{2}$ in.	Cooling	North									
		East									
		South									
		West									
	Heating	—									

APPENDIX C

Tabulation of Data at Times of Heating and Cooling Peak Loads

TABLE C-1—PORTION OF TABULATION OF YEAR ROUND HOURLY HEATING AND COOLING LOADS FOR HOUSE USED AS EXAMPLE. TABLE SHOWING PEAK COOLING LOAD

LOCATION	CONSTRUCTION	YEAR	MONTH	DAY	HOUR	OUTDOOR TEMP.	RADIATION ON HORIZ. SURFACE	WIND (MPH)	INDOOR TEMP.	WALL AND ROOF LOADS	COMPUTING FACTOR	WINDOW LOAD	COMPUTING FACTOR	INFILTRATION LOAD	TOTAL LOAD	MARKER	DAILY TOTAL HEATING LOAD	DAILY TOTAL COOLING LOAD	
1	14B1A	52	9	12	14	89	36.1	10	75	3963	37	13719	11	1847	11399				
1	14B1A	52	9	12	15	89	37.7	8	75	4260	46	16140	11	1971	14241	2			
1	14B1A	52	9	12	16	89	19.1	10	75	4469	30	11836	11	2015	10190				
1	14B1A	52	9	12	17	86	5.8	12	75	4439	20	8339	8	1826	6474				
1	14B1A	52	9	12	18	84	.3	7	75	4119		2421	7	1539	51—				
1	14B1A	52	9	12	19	82		2	75	3671		1883	5	1245	1331—				
1	14B1A	52	9	12	20	81		2	75	3120		1614	5	1035	2361—				
1	14B1A	52	9	12	21	80		2	75	2597		1345	4	890	3298—				
1	14B1A	52	9	12	22	79		3	75	2114		1076	3	723	4217—				
1	14B1A	52	9	12	23	78		4	75	1715		807	2	544	5064—		70, 151—	78, 546	
1	14B1A	52	9	13		77		4	75	1355		538	2	430	5807—				
1	14B1A	52	9	13	1	78		5	75	1102		807	2	388	5833—				
1	14B1A	52	9	13	2	77		6	75	945		538	2	374	6273—				
1	14B1A	52	9	13	3	75		5	75	736				236	7158—				
1	14B1A	52	9	13	4	72		6	75	349		807—	2—	218—	8806—				
1	14B1A	52	9	13	5	72	.2	2	75	124—		807—	2—	314—	9375—				
1	14B1A	52	9	13	6	71	6.7	8	75	536—	9	1345	3—	417—	7738—				
1	14B1A	52	9	13	7	72	23.9	2	75	860—	21	4342	2—	438—	4586—				
1	14B1A	52	9	13	8	79	39.6	2	75	745—	33	9953	3	465	1543				
1	14B1A	52	9	13	9	84	51.5	10	75	15	41	13450	7	791	6126				
1	14B1A	52	9	13	10	87	59.7	12	75	1167	46	15602	9	1251	9890				
1	14B1A	52	9	13	11	88	47.8	11	75	2292	35	12912	10	1588	8662				
1	14B1A	52	9	13	12	88	62.0	13	75	3211	49	16678	11	1831	13590	3			
1	14B1A	52	9	13	13	90	59.7	16	75	3928	53	18292	17	2373	16463	1			
1	14B1A	52	9	13	14	93	48.8	14	75	4533	49	18023	17	2875	17301	1			
1	14B1A	52	9	13	15	93	38.1	16	75	5081	47	17485	21	3325	17761	1	←*PEAK COOLING LOAD*		
1	14B1A	52	9	13	16	93	21.8	15	75	5468	35	14257	19	3559	15154	2			
1	14B1A	52	9	13	17	90	6.7	15	75	5597	23	10222	16	3342	11031				
1	14B1A	52	9	13	18	88	.3	8	75	5424		3497	10	2696	3487				
1	14B1A	52	9	13	19	86		7	75	5003		2959	8	2018	1850				

1	14B1A	52	9	13	20	83		7	75	4372			2152	6	1526	60—		
1	14B1A	52	9	13	21	82		9	75	3673			1883	5	1179	1395—		
1	14B1A	52	9	13	22	81		10	75	3026			1614	5	1003	2487—	62, 930—	
1	14B1A	52	9	13	23	80		12	75	2513			1345	4	880	3392—		122, 858
1	14B1A	52	9	14		78		14	75	2035			807	3	719	4569—		
1	14B1A	52	9	14	1	78		6	75	1660			807	2	544	5179—		
1	14B1A	52	9	14	2	77		8	75	1273			538	2	430	5889—		
1	14B1A	52	9	14	3	76		8	75	965			269	1	321	6575—		
1	14B1A	52	9	14	4	74		8	75	610			269—	1—	231—	8020—		
1	14B1A	52	9	14	5	73	.2	16	75	201			538—	2—	266—	8733—		
1	14B1A	52	9	14	6	72	.8	10	75	187—	1		538—	2—	332—	9187—		
1	14B1A	52	9	14	7	71	1.3	11	75	577—	1		807—	3—	423—	9937—		
1	14B1A	52	9	14	8	69	3.7	10	75	989—	3		807—	5—	643—	10569—		
1	14B1A	52	9	14	9	69	3.9	10	75	1397—	3		807—	5—	827—	11161—		
1	14B1A	52	9	14	10	69	3.3	10	75	1684—	3		807—	5—	893—	11514—		
1	14B1A	52	9	14	11	70	6.3	12	75	1824—	5			4—	850—	10804—		
1	14B1A	52	9	14	12	72	11.5	8	75	1761—	9	1614		2—	648—	8925—		
1	14B1A	52	9	14	13	72	10.6	12	75	1557—	9	1614		2—	468—	8541—		
1	14B1A	52	9	14	14	72	7.3	9	75	1336—	7	1076		2—	402—	8792—		
1	14B1A	52	9	14	15	70	3.8	14	75	1263—	5			5—	579—	9972—		
1	14B1A	52	9	14	16	69	2.9	14	75	1374—	5	269—		6—	866—	10630—		
1	14B1A	52	9	14	17	68	1.3	14	75	1612—	4	807—		7—	1093—	11642—		
1	14B1A	52	9	14	18	67	.2	15	75	1936—		2152—		8—	1294—	13512—		
1	14B1A	52	9	14	19	66		12	75	2281—		2421—		7—	1353—	14185—		
1	14B1A	52	9	14	20	65		12	75	2641—		2690—		8—	1386—	14847—		
1	14B1A	52	9	14	21	64		10	75	2974—		2959—		8—	1448—	15511—		
1	14B1A	52	9	14	22	64		10	75	3267—		2959—		8—	1470—	15826—		
1	14B1A	52	9	14	23	64		8	75	3472—		2959—		8—	1476—	16037—	250, 566—	
1	14B1A	52	9	15		64		10	75	3595—		2959—		8—	1480—	16164—		
1	14B1A	52	9	15	1	63		14	75	3701—		3228—		11—	1681—	16740—		

TABLE C-2—PORTION OF TABULATION OF YEAR ROUND HOURLY HEATING AND COOLING LOADS FOR HOUSE USED AS EXAMPLE. TABLE SHOWING PEAK HEATING LOAD

LOCATION	CONSTRUCTION	YEAR	MONTH	DAY	HOUR	OUTDOOR TEMP.	RADIATION ON HORIZ. SURFACE	WIND (MPH)	INDOOR TEMP.	WALL AND ROOF LOADS	COMPUTING FACTOR	WINDOW LOAD	COMPUTING FACTOR	INFILTRATION LOAD	TOTAL LOAD	MARKER	DAILY TOTAL HEATING LOAD	DAILY TOTAL COOLING LOAD
1	14B1A	52	12	27		34		16	75	13380—		11029—	47—	8665—	41204—			
1	14B1A	52	12	27	1	33		10	75	13568—		11298—	32—	7684—	40680—			
1	14B1A	52	12	27	2	33		14	75	13740—		11298—	40—	7098—	40266—			
1	14B1A	52	12	27	3	32		17	75	13901—		11567—	54—	8212—	41810—			
1	14B1A	52	12	27	4	32		13	75	14073—		11567—	37—	8211—	41981—			
1	14B1A	52	12	27	5	32		15	75	14194—		11567—	45—	7867—	41758—			
1	14B1A	52	12	27	6	32		15	75	14278—		11567—	45—	8171—	42146—			
1	14B1A	52	12	27	7	32	.2	18	75	14317—		11567—	58—	9152—	43166—			
1	14B1A	52	12	27	8	32	6.0	12	75	14344—	22	5649—	32—	8410—	36533—			
1	14B1A	52	12	27	9	34	16.9	11	75	14264—	39	538—	31—	6763—	29695—			
1	14B1A	52	12	27	10	35	27.3	15	75	14048—	54	3766	42—	6826—	25238—			
1	14B1A	52	12	27	11	37	35.5	18	75	13716—	65	7263	51—	8029—	22612—			
1	14B1A	52	17	27	12	37	35.8	24	75	13344—	67	7801	74—	10448—	24121—			
1	14B1A	52	12	27	13	38	31.5	20	75	13005—	69	8608	57—	11415—	23942—			
1	14B1A	52	12	27	14	36	7.9	26	75	12790—	21	4842—	84—	12710—	38472—			
1	14B1A	52	12	27	15	34	6.5	16	75	12859—	29	3228—	47—	12073—	36290—			
1	14B1A	52	12	27	16	33	1.5	20	75	13128—		11298—	65—	11145—	43701—			
1	14B1A	52	12	27	17	30		35	75	13575—		12105—	137—	16503—	50313—	4		
1	14B1A	52	12	27	18	28		36	75	14166—		12643—	148—	22954—	57893—	5		
1	14B1A	52	12	27	19	25		30	75	14897—		13450—	128—	24498—	60975—	6		
1	14B1A	52	12	27	20	22		32	75	15709—		14257—	146—	25284—	63380—	6		
1	14B1A	52	12	27	21	20		30	75	16557—		14795—	140—	26088—	65570—	6		
1	14B1A	52	12	27	22	19		30	75	17349—		15064—	143—	26181—	66724—	6		
1	14B1A	52	12	27	23	17		30	75	18023—		15602—	148—	26676—	68431—	7	1,046,901—	
1	14B1A	52	12	28		17		35	75	18593—		15602—	177—	29095—	71420—	7		
1	14B1A	52	12	28	1	17		30	75	18973—		15602—	148—	29502—	72207—	7		
1	14B1A	52	12	28	2	16		38	75	19220—		15871—	198—	31522—	74743—	7	←*PEAK HEATING LOAD*	
1	14B1A	52	12	28	3	15		30	75	19453—		16140—	153—	31805—	75528—	7		
1	14B1A	52	12	28	4	14		29	75	19717—		16409—	149—	29343—	73599—	7		
1	14B1A	52	12	28	5	14		27	75	19944—		16409—	137—	27359—	71842—	7		
1	14B1A	52	12	28	6	13		26	75	20162—		16678—	133—	25817—	70787—	7		
1	14B1A	52	12	28	7	13	.4	21	75	20373—		16678—	102—	22880—	68061—	7		

1	14B1A	52	12	28	8	13	9.6	19	75	20521—	35	7263—	90—	19472—	55386—	5					
1	14B1A	52	12	28	9	15	22.7	20	75	20525—	53	1883—	93—	17805—	48343—	4					
1	14B1A	52	12	28	10	18	33.7	23	75	20268—	67	2690	105—	18175—	43883—						
1	14B1A	52	12	28	11	21	37.3	20	75	19747—	67	3497	84—	17520—	41900—						
1	14B1A	52	12	28	12	23	36.2	14	75	19043—	68	4304	49—	13897—	36766—						
1	14B1A	52	12	28	13	26	32.3	10	75	18252—	71	5918	37—	9977—	30441—						
1	14B1A	52	12	28	14	27	21.2	13	75	17442—	58	2690	41—	8191—	31073—						
1	14B1A	52	12	28	15	28	9.8	10	75	16756—	45	538	35—	7323—	32747—						
1	14B1A	52	12	28	16	27	1.2	17	75	16296—		12912—	60—	8394—	45732—	4					
1	14B1A	52	12	28	17	27		10	75	16074—		12912—	36—	8510—	45626—	4					
1	14B1A	52	12	28	18	27		14	75	15979—		12912—	46—	8012—	45033—						
1	14B1A	52	12	28	19	26		7	75	16010—		13181—	37—	7723—	45044—						
1	14B1A	52	12	28	20	25		16	75	16150—		13450—	58—	8586—	46316—	4					
1	14B1A	52	12	28	21	24		16	75	16387—		13719—	59—	10001—	48237—	4					
1	14B1A	52	12	28	22	23		13	75	16654—		13988—	44—	9610—	48382—	4					
1	14B1A	52	12	28	23	23		12	75	16920—		13988—	39—	8337—	47375—	4	1, 270, 471—				
1	14B1A	52	12	29		23		6	75	17125—		13988—	39—	7631—	46874—	4					
1	14B1A	52	12	29	1	24		10	75	17208—		13719—	38—	7278—	46335—	4					
1	14B1A	52	12	29	2	23		6	75	17226—		13988—	39—	7159—	46503—	4					
1	14B1A	52	12	29	3	23		8	75	17259—		13988—	39—	7187—	46564—	4					
1	14B1A	52	12	29	4	23		8	75	17284—		13988—	39—	7205—	46607—	4					
1	14B1A	52	12	29	5	23		8	75	17302—		13988—	39—	7211—	46631—	4					
1	14B1A	52	12	29	6	23		5	75	17314—		13988—	39—	7215—	46647—	4					
1	14B1A	52	12	29	7	25	.3	8	75	17261—		13450—	38—	7148—	45989—	4					
1	14B1A	52	12	29	8	26	7.1	5	75	17045—	25	6456—	37—	7005—	38636—						
1	14B1A	52	12	29	9	29	20.5	8	75	16673—	47	269	35—	6767—	31301—						
1	14B1A	52	12	29	10	31	29.3	7	75	16133—	58	3766	33—	6443—	26940—						
1	14B1A	52	12	29	11	33	32.2	11	75	15496—	58	4304	32—	6154—	25476—						
1	14B1A	52	12	29	12	36	29.3	10	75	14772—	55	4304	29—	5707—	24395—						
1	14B1A	52	12	29	13	37	23.0	11	75	14046—	50	3228	29—	5513—	24461—						

DISCUSSION

K. R. SOLVASON, Ottawa, Canada (WRITTEN): The authors are to be complimented on devising a relatively simple method for calculating transient heat flows. The method appears especially useful since it is not necessary to use complicated and expensive computing devices except perhaps for calculation of the response factors. The response factors for the most common enclosing combinations could be tabulated in data books and so enable any air conditioning engineer to make a heat load calculation provided only that he has reliable weather data.

Response factors could perhaps be obtained experimentally to give more reliable values than those calculated. Actual wall sections, for instance, could be subjected to either a step or a cyclical temperature change on one side with heat flow measurements on the opposite side providing the response factors. Such experimental determinations could be made under conditions of temperature and humidity very close to the operating conditions so that any effects such as moisture migration would be included. Many of the wall and roof constructions encountered contain 2 or more parallel heat flow paths of different conductance and capacity through the wall and contain also heat flow paths in the plane of the wall. It is usually very difficult to include more than the conductance normal to the wall in an electrical or hydraulic analogy. Thus a thermal test might in many cases be the most practical method for obtaining response factors.

The authors state, as one of the advantages of their method, that hourly values of actual weather data may be used. The design engineer is, however, seldom interested in calculations based on any particular past weather observations, but must design on the basis of the extremes to be expected in the future. Rather than avoiding the selection or assumption of outdoor conditions the method perhaps indicates more clearly the need for realistic design data which can be used for transient heat load calculations. There is perhaps considerable merit in having design data available on an hourly basis since hourly observations of previous weather can most conveniently be used in arriving at design figures.

DR. S. F. GILMAN, Syracuse, N. Y.: This paper presents another method of thermal circuit analysis to add to the steadily growing list. It seems to me that the method is best suited for 1) equipment operating cost studies and 2) investigations of the effect of variable indoor air temperature.

The first results from the emphasis in the paper on the use of actual weather data, as well as the ability, through computing machines, to calculate the heating and cooling load for each hour of a year or more. Comparisons can thus be made of computed versus actual operating times and energy costs. As for cooling load calculations made to select equipment, is it still not necessary to select some sort of anticipated outdoor conditions that we normally call *design conditions*? For any region, a particular cycle of outdoor temperature and solar radiation must be selected and all predicted loads based on this. Can Conclusion 5 actually be implemented?

The second item, variable indoor air temperature, is an important effect about which extremely little information is available. Here I believe the method can contribute a good deal. As to response factors with constant room air temperature, the equivalent has already been done; e.g., data are given in Tables 9 and 10 of Chapter 13 of THE GUIDE. Is not the bracketed term in Equation 6 simply the $U\Delta t$ of the Equivalent Temperature Differential Method already in THE GUIDE?

The variable room air temperature condition is very difficult to validate by comparing a network solution with experimental data. Data we have collected in a residential test facility has so far been inconclusive. The troublesome spots are picking values for the circuit constants. For example, the *furniture and partitions* branch in Fig. 8 lumps all furnishings into one resistor and one capacitor. Some typical questions we ran into are: "What is the average surface film coefficient of a chair?" and "What value do you use for some sort of average thermal resistance between the surface and the point at which the physical mass is presumed to be concentrated?".

Even for the constant room temperature case, there are difficulties. Neglect of radiation exchange between walls, ceilings and floors; as has been done here, causes an error of unknown amount. Whether the film coefficients of convection can be taken as constants is not known. The big unanswered question is: Does the network solution with all its assumptions correctly predict the actual physical system performance? For example, when applied to a particular residence, to what degree of accuracy does the solution predict the actual measured peak cooling load? Once validated, calculations can be made for any structure under widely varying conditions.

These aspects were the subject of considerable discussion at our TAC on Thermal Circuits meeting yesterday. As a result, a research program was formulated which we hope will eventually give us the answers that are so badly needed.

In summary, the *thermal response* method has considerable potential but complete validation of its *practical use* must await improved knowledge of the values that have to be assigned to the various circuit elements.

H. B. NOTTAGE, Encino, Calif.: May I speak partially on behalf of the cooperative project on thermal circuits at the University of California, being carried out by Mr. Harry Buchberg?

It is unfortunate that this interesting paper did not clear thru the TAC on Thermal Circuits, where valuable discussion could have been obtained in advance.

The author's work is but one aspect of general circuit analysis using the Laplace Transform technique. Use of this technique has been under study at UCLA for something over a year, but a paper on it has not yet been released, pending the accumulation of adequate *new* results.

The authors assume a certain familiarity with electrical language. The *transfer heat admittance* is one means of designating an equivalent single-admittance (or single-impedance) circuit beyond the *control point* of heat input. This sounds simple, and perhaps it is relatively so for the simple house with linear passive circuit elements studied in this paper.

In the more general case, such as is being studied at UCLA, the structure is more complicated, variable heat sources within the building are considered, and the need becomes to establish circuit-branch impedances (or admittances) for representative structural sections or groups of sections. These can be tabulated, and then used to develop *families* of circuit solutions. In the language of the authors, any established circuit has a *transfer heat admittance*,—which may not be constant when non-linearities and other internal sources are involved.

The following questions are posed concerning the authors' method, as presented.

1. How does one establish temperatures at significant points within the structure?

2. How can one establish thermal currents at significant points within the structure?

3. How consider glass sections with radiation transmitted directly?

4. In general, how do the authors consider internal radiation exchange and reflections?

5. How do they treat equipment operating characteristics when the internal air temperature cycles?

6. How can one treat internal sources?

7. How would they introduce automatic controls?

8. Noting that areas are involved in circuit-branch resistances and capacities, how would one apply this method to another 1800 sq ft house but with quite different geometrical proportions?

9. What would be the effect of changes in the compass orientation of a house?

10. Please explain further the *sun* arrows shown in Fig. 8?

11. Can one pick off the magnitude and time of the peak instantaneous load without computing a complete cycle?

12. How about transient and aperiodic inputs?

Answers to questions such as the foregoing may help to define the limits of the method presented.

In conclusion, the authors are to be commended for having published an interesting step in the general domain of *system* analyses.

C. T. DERGEL*, Lynn, Mass.: I think I better clear the air a little bit. I am at Poly but I am not here representing Poly. I am here from the General Electric Company. I was with the General Electric Company at the start of this study, which began about 10 years ago. There I think we have to say, in this sort of a first phase, that the mission was not accomplished. It was because it was poorly defined, but it wound up as a company report and then disappeared in the file.

Sometime later, with a lot more clear thinking behind us, we started again. This was about 5 years ago, and the paper was published two years ago and presented in Swampscott.

If you try to make these calculations, by whatever method, I think you will find you have your hands full. To arrive at a circuit is a terrific problem. One of the first failures was the fact that we couldn't arrive at a circuit.

So, on the second phase we just took the bull by the horns and took constants. Maybe they aren't good; maybe they aren't the same all over the house, but we needed to do something.

We wanted to know thermal storages and didn't even know specific heats. So, even if you are willing to say that the specific heat is constant in the wall, you don't even know what it is. So, I agree that finding a circuit is a difficult task.

However, we did it, or it was done. The calculation method turned out to be formidable and I think that these authors have made a terrific contribution to the calculation method, which is all they claim, and I am sure that all who are interested in thermal storage effects will find some information out of applying this method. I want to commend the authors on what they have done.

R. A. GONZALEZ, (Dayton, Ohio): I don't feel qualified to judge the method itself, but I believe that the method is completely valid for a heating study and can be developed directly as presented for heating use, because we must be able to heat the residence when the occupants go away for an evening or perhaps a complete day.

On the cooling side, the method as currently presented does not appear to me to be complete because it does not take into account the internal generated heat of the average high or maximum occupany. The suppertime load of usage in a home coincides frequently with the maximum heat delivery to the equipment, and this internal heating load, while it has not been thoroughly investigated, appears to be in the order of 6,000 to 12,000 Btu per hour for the average home, and that quantity can very readily upset all of these calculations and all of these temperature expectations based on integrated calculations that take other items into account but do not include any consideration of the internally generated heat loads.

So I feel that the method is very usable, for heating, as presented, but for cooling it will have to incorporate the data on the internally-generated load.

P. R. ACHENBACH, Bethesda, Md.: I would like to make a comment as a member of the TAC on Weather Data. This method probably makes the job of the Weather Data Committee tougher instead of easier.

Whereas now we are selecting and have listed in THE GUIDE what we call design temperatures for use in steady state equations, if we use this method we would have to replace that with a typical or a design curve as shown here in Fig. 3. Along with that we would probably also like to have a design curve for wind velocity.

I am sure that if we can arrive at that stage, the service we will be doing the design engineer and the consulting engineer who wants to accurately take into account the heat

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* General Electric Company

capacity, and other factors, of a structure, will be improved. But there is also a job still left for the Weather Data Committee in picking out the right transient pattern to use for any of these methods.

Right now the Weather Data Committee is trying to evolve a set of design data that will provide equal risks all over the country of being underheated or overheated to improve the present data that is in THE GUIDE. This is still a way off as far as weather data selection is concerned.

AUTHORS' CLOSURE (W. R. Brisken and S. G. Reque): We wish to thank the discussers for the interest which they demonstrated. The many favorable comments are appreciated greatly.

Thermal circuit techniques have been used many years for solving a variety of transient heat flow problems. One sector of this wide area is calculating of heating and cooling loads for buildings. While we are observing carefully every advance made in this field, especially the research work conducted under guidance of the TAC on Thermal Circuits, we nevertheless were confronted with the immediate need for a simple, highly accurate method of load calculating which would permit us to solve the problems of transient heat flow which are ever present in design and application engineering of year-round air conditioning equipment.

We strongly believe that the Thermal Response Method meets the requirement. It differs from other methods insofar as it applies the convolution technique. The actual problem is divided into 2 parts. The first is to establish a characteristic series of numbers which is the Transfer Heat Admittance and which needs to be calculated or determined by experiment only once for each type of structure. The second is to utilize this characteristic series of numbers for heat load calculations wherever the particular structure is involved. The first part can be performed on a computer most suitable for such calculations and may be in part or entirely verified by experiments. The response factors for the most common structures may then be published in handbooks. The second part eliminates the need for computers and makes it possible for everyone to calculate transient heat flows for such structures or combinations of structures for which response factors are available.

Outdoor weather conditions must be selected for every load calculating method. However, it should be recognized that the *combination* of effects (outdoor temperature, solar radiation and wind) produces an extreme condition. This combination affects different buildings in a different way. There is considerable evidence in our studies of thermal circuits that extreme loads are a function of structure as well as of the external conditions. It will be necessary to study the problem of design weather conditions from this viewpoint. The thermal response method can be of great value to such studies since weather data recorded by the U. S. Weather Bureau on business machine cards can be used for study of extreme conditions. We wish conclusion 5 to be understood in this way that an application method as discussed in our paper would be based on weather conditions which have actually existed during a period of, say, 10 years prior to date and which have produced extreme loads. These conditions would be incorporated in *Load Factors*, which means that the need for selecting design temperatures would be eliminated.

Response factors as described in the paper are a series of numbers characterizing the flow of heat from a structure as a result of a temperature pulse. They are not the same as Equivalent Temperature Differentials to which Dr. Gilman refers. However, the Load Factors which can be derived from thermal response calculations and which are suggested for a new application method are indeed expressions of $U\Delta t$. But it is proposed that these load factors be calculated from the worst *weather cycles* experienced at different locations and not from a combination of design temperature and solar radiation.

We do agree with Dr. Gilman that it is troublesome to pick circuit constants for furniture and partitions. But for this particular problem the thermal response method offers unique possibilities through its basic approach. Rather than trying to select circuit

constants—which is practically impossible—it may be preferred in such cases to determine response factors by experiment.

For our first presentation of the thermal response method we purposely chose a drastically simplified circuit (Fig. 8). We intended to acquaint the members of the ASHAE with the new concept and its usefulness to heat load calculations. Most of Dr. Nottage's questions fall into an area which is outside the *intended* scope of our paper because they concern either circuit complications or *systems* analysis including equipment characteristics and automatic controls. The thermal response method has been found by our company to be very useful indeed for systems studies. But to answer Dr. Nottage's questions in detail would be attempting a new paper in this limited space.

We should repeat that the purpose of presenting this paper was to acquaint the members of the ASHAE with the thermal response method. It was not because of the authors' action that the paper did not clear thru the TAC on Thermal Circuits. However, we are surprised at the opinion that the paper should *clear* through a TAC prior to presentation, when the paper had not been prepared on the initiative of the TAC.

With respect to Mr. Gonzalez' question we feel that internal loads from appliances or occupancy can be handled as dc loads. It may, however, be advisable to apply them with a time constant as has been done in the circuit branch for infiltration. The question if the thermal response method can be applied as well to cooling as to heating studies really touches another aspect of load calculations. This paper was confined to sensible loads only. But it appears from our present studies that the same technique can be applied to latent loads also. It may be said that the real value of a heat load calculating method is established only if sensible as well as latent loads can be determined with satisfactory accuracy.

It will always be difficult to decide to what degree individual details must be included in load calculations. We feel that the error which may result from simplifications should not be expressed in terms of load but rather in degrees deviation from design temperature. The thermal response method can be used advantageously for such calculations as has been demonstrated in Fig. 13 of this paper.

1580

No. 1580

AIR VELOCITIES IN TWO PARALLEL VENTILATING JETS

By ALFRED KOESTEL* AND J. B. AUSTIN, JR.**, CLEVELAND, OHIO

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS in cooperation with Case Institute of Technology, Cleveland, Ohio

IN THE DESIGN of air distribution outlets the factors of primary concern are the air flow pattern, or cross-sectional velocity distribution, and the maximum velocities encountered at a given distance from the nozzles. When analyzing the behavior of 2 relatively close-spaced jets it is worthwhile to have equations by which the maximum velocities may be estimated. In addition, it is desirable to establish principles by which equations may be derived for determining velocities anywhere in the jet stream.

The problem is one of interference, considering one jet as interfering with another. The principle involved is that of one jet superimposing upon another, whereby the respective momenta become additive in accordance with the laws of momentum.†

Equations with experimentally evaluated constants are presented in this paper in a form such that a designer of air distribution systems may estimate the maximum jet velocities at a given distance from the jet nozzle, or such that he may estimate the jet throw when applying an arbitrary terminal velocity.

The detailed mathematical analyses appear in the Appendix.

TWIN PARALLEL AIR JETS

An air jet discharged from a nozzle will have a cross-sectional velocity profile which can be approximated by a normal probability curve. When 2 parallel air jets interfere with each other the resulting velocity profile is as shown in Fig. 1. In each jet there is a constant outlet velocity core, and the jet behaves independently until a point is reached at which the 2 interfere. In Fig. 1, "A" is the point of interference, and its distance out from the nozzles varies with the distance between the nozzles. From the point of interference on out, the 2 jets may be considered

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† Similar work on twin jet interference was performed by the University of Illinois at about the same time as it was at Case Institute of Technology in 1952. The jet investigations made at Case covered greater distances from the nozzles and lower nozzle outlet velocities. The resulting equations differed from those of the University of Illinois in form, but were based upon the same principle of superposition.

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as superimposing or interfering, and the velocity profile is the result of their combined momenta.

From the nozzles to the point of interference, the maximum velocity, as for a single jet, will be on the centerline of each jet. Then the velocity on a line midway between and parallel to the two jet centerlines increases until it equals the jet centerline velocity. The distance from the nozzles to this point is X_i. From this point on, the maximum velocity of the combined jet stream is on the midway line, and the profile is almost as though emanating from a single nozzle of twice the area of one of the two nozzles.

EQUATIONS AND THEIR APPLICATIONS

Applying the principle of conservation and addition of momentum, or superposition, the centerline velocity of a single jet in the presence of another twin jet

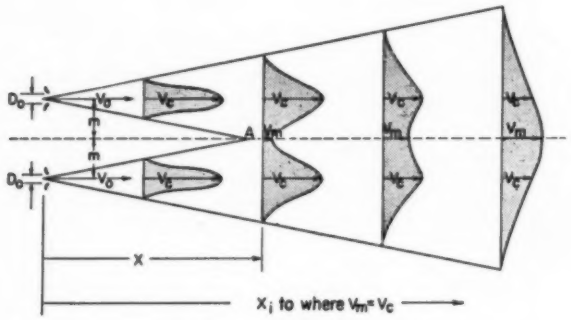

FIG. 1—DIAGRAM OF TWO PARALLEL JETS SHOWING VELOCITY PROFILES PRODUCED BY INTERFERENCE

can be found by Equation 1

$$\frac{V_c}{V_0} = \frac{K}{X/D_0}\left[1 + e^{-16(Km/X)^2}\right]^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The symbols are defined in the Nomenclature.

The velocity on the line midway between the 2 jet centerlines may be determined by Equation 2

$$\frac{V_m}{V_0} = \frac{K\sqrt{2}}{X/D_0}\, e^{-2(Km/X)^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

In determining the maximum velocity in a twin jet stream at a given distance, X, from the nozzles, both the above equations may be solved to see which yields the greater value of velocity. The alternative method is to solve for X_i by means of the equation,

$$X_i = 2.56\, Km \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Equation 3 gives the value for X_1 at which distance V_e equals V_m. If the distance in question is less than X_1, the maximum velocity may be found by Equation 1. If X is greater than X_1, the maximum velocity will be on the midway line and may be found by using Equation 2.

At large values of X/m the exponential term of each, Equations 1 and 2, is seen to approach a value of one, and both equations become,

$$\frac{V_e}{V_0} = \frac{V_m}{V_0} = \frac{K\sqrt{2}}{X/D_0} \quad \ldots \ldots \ldots \ldots \quad (4)$$

This is also the equation for the maximum centerline velocity of a single jet when the jet area is doubled. In addition, Equation 4 applies when a single outlet is located adjacent to a wall or ceiling, and the air is discharged along the surface.[1]

Thus, in using Equation 1 or 2 to determine the maximum velocity in a twin jet stream at a given distance from the nozzles, it is necessary to know the nozzle diameters, the distance between the nozzle centerlines, the outlet velocity, and some value for K. At large values of X, the distance between the nozzle centerlines is insignificant, and the jet behaves as a single jet for which Equation 4 then applies.

The value of K to be used apparently varies with outlet size and velocity.[1] It has been found from this investigation that, for an outlet velocity of 3000 fpm through a 1-in. nozzle, the equations agree well with the test data when K equals 4.5.

For outlet velocities of 23,000 fpm from nozzles of similar size,[2] a K of 6.5 gives better test agreement with Equations 1 and 2. There are experimental indications that K is a function of the Reynolds number evaluated at the discharge nozzle only when the Reynolds number is low. For a Reynolds number greater than about 3×10^4, values of K will be greater than 6.0. Values of K can be expected to lie between 6.5 and 4.0 where the lower values are for small size outlets and low velocities.

SIMPLIFYING ASSUMPTIONS

Assumptions underlying Equations 1 and 2 are as follows:

1. Outlet velocities from each of the twin jets are equal.
2. No external forces act upon the jets other than the pressure behind the nozzle; *i.e.*, they are considered free except for the interference of one with the other.
3. The absolute value of the air density is constant throughout the jets.
4. The angle of expansion for the velocity profiles is constant.
5. The apparent point source of each jet lies on the face of the nozzle outlet. This may not be exactly true and will only affect results close to the nozzle.

COMPARISON WITH TEST DATA

Fig. 2 is a plot of Equation 1, with a K of 6.5 for different spacings between nozzles. For a nozzle spacing of 8 diameters and a distance out from the nozzles of 100 diameters, the velocity on the jet centerline is just slightly greater than for a single jet without interference. At the same distance out and at a nozzle spacing of 3 diameters, V_e is 28 percent greater than for a single jet. The velocities V_e

[1] Exponent numerals refer to References.

or V_m for a twin jet can never be greater than 1.414 times the velocity V_c of a single jet when the contrast is made on the basis of the same outlet velocity, V_0, and the same size, D_0. Equation 4 differs from the equation for a single jet only by the factor, $\sqrt{2}$, as shown by Equation 5 for a single jet.

$$V_c/V_0 = K/(X/D_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The curves of Fig. 2 originate at $X/D_0 = 6.5$ and $V_c/V_0 = 1$, since from Equation 5 at $V_c/V_0 = 1$, $X/D_0 = K$. This distance represents the approximate length of the constant velocity core of the jet.

Fig. 3 is a plot of Equation 2 similar to Fig. 2. As X becomes infinitely large in both Figs. 2 and 3, the family of curves tends to coincide and become asymptotic along the X-axis.

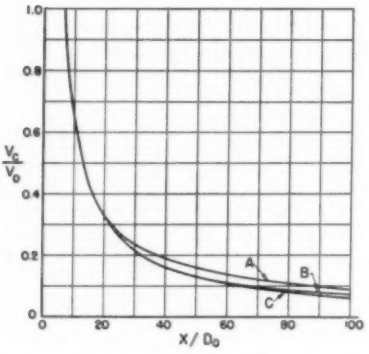

FIG. 2—PLOT OF EQUATION 1 FOR VARIOUS DISTANCES BETWEEN TWO NOZZLES WITH K EQUAL TO 6.5; A IS $2m/D_0 = 3$ (TWIN PARALLEL JETS); B IS $2m/D_0 = 8$ (TWIN PARALLEL JETS); C IS SINGLE JET

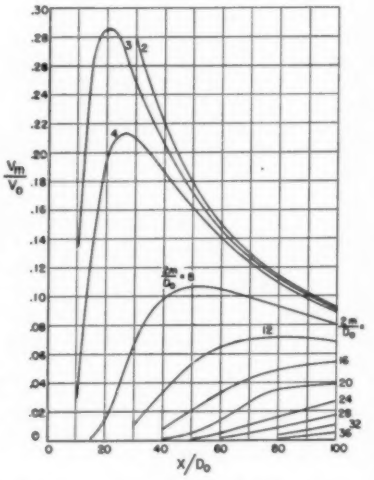

FIG. 3—PLOT OF EQUATION 2 FOR VARIOUS DISTANCES BETWEEN TWIN PARALLEL NOZZLES WITH K EQUAL TO 6.5

Tests conducted at Case Institute of Technology were made on two 1-in. diameter nozzles with a spacing of 5 in., and the results are plotted in Figs. 4 and 5. The curves of Figs. 4 and 5 are based on a K value of 4.5, and the data were obtained at a nozzle velocity of about 3000 fpm. A map of the field velocity distribution covering a distance out to 100 diameters is shown for a typical test in Fig. 6. Velocity measurements were made with a calibrated bridled vane anemometer.

Test results obtained at the University of Illinois[2] are shown in Figs. 7 and 8, where the jet centerline velocities and the velocities at the midway line are compared with Equation 2 for a K equal to 6.5. Measurements were made on two 0.898-in. diameter nozzles spaced 3 in. apart, and velocity traverses were made out

to a distance of 30 diameters. The outlet velocity was 23,000 fpm. Measurements were made with a total-head impact tube.

PRACTICAL EXAMPLE

As a simple example in finding the magnitude of the jet velocities produced by 2 nozzles spaced in parallel, consider two 1-ft diameter nozzles with a centerline spacing of 2 ft discharging isothermal air at a velocity of 1000 fpm. It is required to find the maximum jet velocity at (*a*) a distance of 10 ft from the nozzles and (*b*) at a distance of 20 ft from the nozzles.

Assuming a K equal to 6.5, the equi-velocity distance out from the nozzle, according to Equation 3, is

$$X_1 = 2.56 \times 6.5 \times 1 = 16.7 \text{ ft}$$

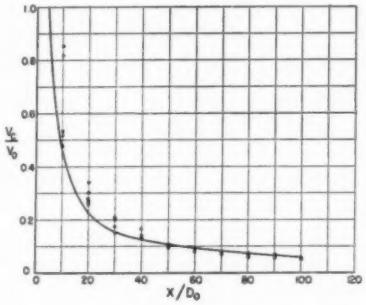

FIG. 4—COMPARISON BETWEEN EQUATION 1 (CURVE) WITH K EQUAL TO 4.5 AND DATA (CIRCLES) FROM CASE INSTITUTE OF TECHNOLOGY. DATA:—V_0 = 3000 FPM, 5 IN. BETWEEN NOZZLE CENTERLINES, TWO 1-IN. NOZZLES

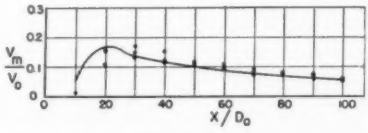

FIG. 5—COMPARISON BETWEEN EQUATION 2 (CURVE) WITH K EQUAL TO 4.5 AND DATA (CIRCLES) FROM CASE INSTITUTE OF TECHNOLOGY. DATA:—V_0 = 3000 FPM, 5 IN. BETWEEN NOZZLE CENTERLINES, TWO 1-IN. NOZZLES

Therefore, at a distance of 10 ft, which is less than the equi-velocity distance of 16.7, the maximum jet velocity can be computed from Equation 1:

Solution (*a*)

$$V_c = 1000 \times \frac{6.5}{10/1} \{1 + e^{-16[(6.5 \times 1)/10]^2}\}^{\frac{1}{2}}$$

$$V_c = 1000 \times [6.5/(10/1)] \times (1.00114)^{\frac{1}{2}} = 651 \text{ fpm}$$

At a distance of 20 ft, which is greater than the equi-velocity distance of 16.7, the maximum jet velocity can be computed from Equation 2:

Solution (*b*)

$$V_m = \frac{1000 \times 6.5\sqrt{2}}{20/1} e^{-2[(6.5 \times 1)/20]^2}$$

$$V_m = [1000 \times 6.5/(20/1)] \times 1.145 = 372 \text{ fpm}$$

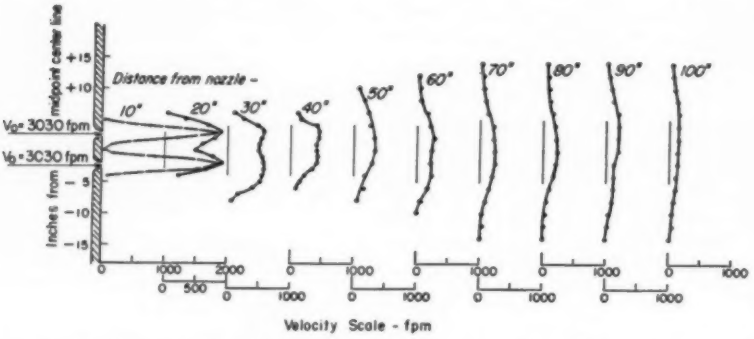

FIG. 6—MEASURED VELOCITY DISTRIBUTION IN A TWIN JET FIELD. TWO 1-IN. NOZZLES, 5 IN. BETWEEN NOZZLE CENTERLINES. DATA FROM CASE INSTITUTE OF TECHNOLOGY

Equation 4 could have been applied only for large distances from the outlet or if it were assumed that the two nozzles are so closely spaced so as to behave as a single outlet.

CONCLUSIONS AND RECOMMENDATIONS

The equations presented in this paper permit a designer of air distribution systems to determine the maximum velocity in a jet stream issuing from two parallel nozzles. The method of deriving the equations is included in the Appendix. This principle of adding momenta, or superposition of interfering jets, can be applied similarly but with more difficulty to more than two jets. Detailed equations by

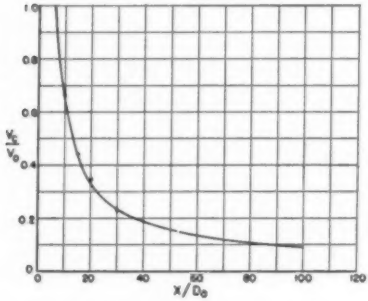

FIG. 7—COMPARISON BETWEEN EQUATION 1 (CURVE) WITH K EQUAL TO 6.5 AND DATA (CIRCLES) FROM THE UNIVERSITY OF ILLINOIS. DATA:—V_0 = 23,000 FPM, 3 IN. BETWEEN NOZZLE CENTERLINES, TWO 0.898-IN. NOZZLES

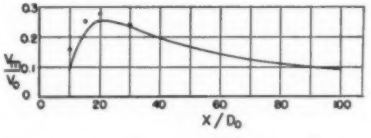

FIG. 8—COMPARISON BETWEEN EQUATION 2 (CURVE) WITH K EQUAL TO 6.5 AND DATA (CIRCLES) FROM THE UNIVERSITY OF ILLINOIS. DATA:—V_0 = 23,000 FPM, K = 6.5, 3 IN. BETWEEN NOZZLE CENTERLINES, TWO 0.898-IN. NOZZLES

which the entire jet velocity profile can be defined are not presented in this paper; the basic equations presented in the Appendix can be manipulated to formulate velocity profile equations.

When relatively large-sized outlets are encountered in air-conditioning practice, it is recommended that a K value of 6.5 be used in Equations 1 ,2 and 4. Hence, if the outlet in question happens to have an exceptionally low discharge velocity, the velocities predicted from Equations 1, 2 and 4 will be on the safe side, higher than the actual jet velocities.

The validation of Equations 1 and 2 rests upon test results obtained from two twin-nozzle configurations; namely, two 0.898-in. diameter nozzles spaced 3 in. apart, and two 1-in. diameter nozzles spaced 5 in. apart. It is recommended that additional data be obtained covering a greater range of outlet sizes, dimensions, and spacings.

Care should be taken when applying the equations to practical-type outlets which have discharge velocity patterns that are different from those of the idealized conditions of the experiment. Deviations can be expected between the jet velocities computed from the equations and the actual velocities when these are below about 150 fpm; the actual velocities will be less than those computed by formula.

ACKNOWLEDGMENTS

The authors are indebted to Prof. G. L. Tuve for his help and suggestions. The experimental work was performed by J. J. Reinmann, student in mechanical engineering. The figures were prepared by Mrs. Ruth Strahosky.

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2. Transport of Momentum, Mass, and Heat in Turbulent Jets, by L. G. Alexander, T. Baron, E. W. Comings (University of Illinois Engineering Experiment Station, *Bulletin Series No.* 413).

NOMENCLATURE

V_0 = outlet velocity (at vena contracta), feet per second.
V_c = jet velocity along the centerlines through the nozzles, feet per second.
V_m = jet velocity along a line midway between the two nozzles, feet per second.
X = distance from apparent point source (assumed to lie on the face of the outlet), feet.
$2m$ = distance between the two nozzles measured from the nozzle centers, feet.
X_i = equi-velocity distance or the distance X at which V_m equals V_c, feet.
D_0 = diameter of nozzle, feet.
K = constant, length in diameters of the constant velocity core.
e = the base for the Napierian system of logarithms $e = 2.718$.

APPENDIX

Dual Jet Interference Analysis

Three Design Relations Are Developed:

1. Velocity midway between the two jets (V_m) in terms of the nozzle exit velocity (V_0), the distance between the two nozzles ($2m$), and the distance out from the nozzles (X).

2. Velocity on the centerline of a nozzle (V_c) in terms of the nozzle exit velocity (V_0), the distance between the 2 nozzles ($2m$), and the distance out from the nozzles (X).
3. The equi-velocity distance out from the nozzles (X_1, at which point V_m equals V_c) in terms of the distance between the nozzles ($2m$).

I. Velocity Midway Between the Two Jets: Assuming a normal probability type curve for the velocity profile of each jet,

$$V = V_c e^{-ar^2}$$

where a is a shape factor, and V is the velocity at distance, X, and radius, r, from the jet centerline.

To obtain the momentum flux profile, square each side of the above equation and multiply by the mass density:

$$\rho V^2 = \rho V_c^2 e^{-2ar^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (A\text{-}1)$$

The total momentum in each jet is

$$M_x = \Sigma \rho (A\,V) V = \Sigma \rho A\, V^2$$

$$A = 2\pi r\, dr$$

$$M_x = \int_{r=0}^{r=\infty} 2\pi \rho V^2 r\, dr$$

Substitute the velocity profile equation for V:

$$M_x = 2\pi \rho V_c^2 \int_{r=0}^{r=\infty} e^{-2ar^2} r\, dr$$

and integrating,

$$M_x = \frac{\pi \rho V_c^2}{2a}$$

The momentum of a single jet at the nozzle exit is

$$M_0 = \rho A_0 V_0^2 = \rho \frac{\pi D_0^2}{4} V_0^2$$

Assuming conservation of momentum,

$$M_0 = M_x$$

Then

$$\rho \frac{\pi D_0^2}{4} V_0 = \frac{\pi \rho V_c^2}{2a}$$

Rearranging terms:

$$\frac{a}{2} D_0^2 \rho V_0 = \rho V_c^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (A\text{-}2)$$

Substituting Equation A-2 into Equation A-1, we have,

$$\rho V^2 = \frac{a}{2} D_0^2 \rho V_0 e^{-2ar^2}$$

The momentum flux at $r = m$ per jet is

$$\rho (V_m')^2 = \frac{a}{2} D_0^2 \rho V_0 e^{-2am^2}$$

Assuming that the total momentum flux midway between the 2 jets is the sum of the momentum flux from each jet (at the point of interference),

$$\rho V_m^2 = 2\left(\frac{a}{2} D_0^2 \rho V_0^2 e^{-2am^2}\right)$$

or

$$\left(\frac{V_m}{V_0}\right)^2 = aD_0^2 e^{-2am^2}.$$

$$a = \frac{C_2}{X^2}, \text{ a shape factor,}$$

and

$$C_2 = 2K^2$$

where K is the length of the constant velocity zone in diameters and is measured from the apparent point source which is assumed to lie on the face of the outlet. K is about 6.5 and is determined experimentally.

$a = \frac{C_2}{X^2}$ is determined by assuming a constant angle of expansion, α_v, for the velocity

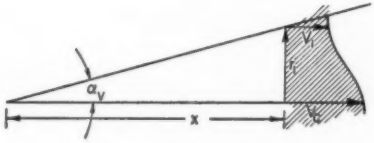

FIG. A-1—JET EXPANSION

profile. Then the ratio V_1/V_c measured along the expansion line becomes independent of X.

From Equation A-1,

$$\frac{V_1}{V_c} = e^{-ar_1^2}$$

From the sketch, $\tan \alpha_v = \frac{r_1}{X}$. See Fig. A-1.

$$\frac{V_1}{V_c} = e^{-aX^2 \operatorname{Tan}^2 \alpha_v}$$

Since $\frac{V_1}{V_c}$ is constant,

$$\frac{V_1}{V_c} = e^{-aX^2 \operatorname{Tan}^2 \alpha_v}$$

$$\ln\left(\frac{V_1}{V_c}\right) = -aX^2 \operatorname{Tan}^2 \alpha_v = \text{constant}$$

and $\operatorname{Tan}^2 \alpha_v$ is constant.

Therefore,

$$aX^2 = C_2$$

or

$$a = \frac{C_2}{X^2}$$

Substituting $a = \frac{C_2}{X^2}$ and $C_2 = 2K^2$ into the equation

$$\left(\frac{V_m}{V_0}\right)^2 = aD_0^2 e^{-2am^2}$$

$$\left(\frac{V_m}{V_0}\right)^2 = \frac{C_2}{X^2} D_0^2 e^{-(4K^2m^2)/X^2}$$

$$\left(\frac{V_m}{V_0}\right) = \frac{\sqrt{2}}{X} KD_0 e^{-(2K^2m^2)/X^2} \qquad \text{[Equation 2 of text]}$$

Rearranging terms:

$$\left(\frac{V_m}{V_0}\right) = \frac{K\sqrt{2}}{\frac{X}{D_0} e^{2(Km/X)^2}} \quad \ldots\ldots\ldots \quad \text{(A-3)}$$

K has a value which is approximately equal to 6.5. At large values of X at which the 2 jets have coalesced, the quantity $e^{2(Km/X)^2}$ approaches unity, and Equation A-3 becomes

$$\frac{V_m}{V_0} = \frac{K\sqrt{2}}{X/D_0}, \qquad \text{[which is Equation (4) of paper.]}$$

II. Velocity on the Centerline of a Nozzle: The momentum on the centerline of one of two parallel nozzles is the sum of the momentums of each jet. The momentum of the jet from the nozzle on the centerline in question is

$$\rho(V_c')^2 = \rho V_0^2 D_0^2 \frac{a}{2}$$

The momentum of the jet from the other nozzle is figured with V at $r = 2m$, or

$$\rho(V_c'')^2 = \rho V_0^2 D_0^2 \frac{a}{2} e^{-2a(2m)^2}$$

The sum of the 2 momenta is

$$\rho V_c^2 = \rho V_0^2 D_0^2 \frac{a}{2} + \rho V_0^2 D_0^2 \frac{a}{2} e^{-2a(2m)^2}$$

$$= \rho V_0^2 D_0^2 \frac{a}{2} [1 + e^{-8am^2}]$$

Substituting $\frac{2K^2}{X^2}$ for a, and cancelling ρ,

$$V_c^2 = V_0^2 D_0^2 \frac{2K^2}{2X^2} [1 + e^{-8((2K^2)/X^2)m^2}]$$

$$\left(\frac{V_c}{V_0}\right)^2 = \left(\frac{K}{\frac{X}{D_0}}\right)^2 [1 + e^{-16((K^2m^2)/X^2)}]$$

$$\frac{V_c}{V_0} = \frac{K}{\frac{X}{D_0}} [1 + e^{-16(Km/X)^2}]^{\frac{1}{2}} \quad \ldots\ldots\ldots \quad \text{(A-4)}$$

and which is the same as Equation 1 of the paper.

If m is small, and X is large, $e^{-16(Km/X)^2}$ approaches 1, and

$$\frac{V_c}{V_0} = \frac{K\sqrt{2}}{X/D_0}$$

which is the same as Equation 4 of the paper.

III. The Equi-Velocity Distance Out from the Nozzles (X_i, at which Point $V_m = V_c$) in Terms of the Distance Between the Nozzles ($2m$): To find the value of X at which $V_c = V_m$, Equations A-3 and A-4 are set equal to each other.

$$\frac{V_m}{V_0} = \frac{V_c}{V_0} = \frac{K\sqrt{2}}{X_i/D_0}\, e^{-2[(Km)/X_i]^2} = \frac{K}{X_i/D_0}\,\{1 + e^{-16[(Km)/X_i]^2}\}^{\frac{1}{2}}$$

Squaring both sides and cancelling,

$$2e^{-4[(Km)/X_i]^2} = 1 + e^{-16[(Km)/X_i]^2}$$

Solving by trial and error,

$$\frac{K_m}{X_i} = 0.391$$

or

$$X_i = \frac{2mK}{0.782} \quad \ldots\ldots\ldots\ldots \quad \text{(A-5)}$$

Simplifying Equation A-5, gives $X_i = 2.56\ Km$, which is Equation 3 of the paper.

APPENDIX NOMENCLATURE

V_0 = Outlet velocity (at vena contracta), feet per second.
V_c = Jet velocity along the centerlines through the nozzles, feet per second.
V_m = Jet velocity along the line midway between the 2 nozzles, feet per second.
X = Distance from apparent point source (assumed to lie on the face of the outlet), feet.
r = Radius, feet.
V = Jet velocity at radius, r, at distance X, feet per second.
$2m$ = Distance between the 2 nozzles measured from the nozzle centers, feet.
X_i = Equi-velocity distance or the distance X at which V_m equals V_c, feet.
D_0 = Diameter of nozzle, feet.
K = Constant, length in diameters of the constant velocity core.
ρ = Air density, slugs per cubic foot.
a = A shape factor.
C_2 = A constant.
α_v = Angle of expansion of jet velocity profile, degrees.
$\rho(V_c')^2$ = Momentum on jet centerline due to nozzle on that centerline, pounds per square foot.
$\rho(V_c'')^2$ = Momentum on jet centerline due to the twin nozzle parallel to the centerline, pounds per square foot.
e = The base for the Napierian system of logarithms e = 2.718.
M_0 = Total jet momentum evaluated at the nozzle, pounds.
M_x = Total jet momentum at distance X, pounds.

DISCUSSION

H. B. NOTTAGE, Encino, Calif. (WRITTEN): In accord with the postulates advanced in this paper, and which postulates are in common use, a consequence for single jets is

that the total mass flow in the jet increases linearly with axial distance from the source. This also means that the rate of entrainment flow is the same for each unit of axial length.

Now, when 2 jets overlap it would be expected that entrainment would be markedly interfered with in the region of interaction. Also, direct superposition would not, in general, over a *wide range of conditions*, be expected to replace entrainment. Yet, the test results follow the predictions from superposition. What might this denote as to limits of validity?

LINN HELANDER, Manhattan, Kan.: It would be interesting to see a plot of mass rate flow against travel for the idealized twin-jets. For large values of X/m, the velocity distribution and therefore the mass rate of flow of the idealized twin jets would seem to approach that of a single jet formed by consolidating the twin jets into a single-outlet jet, just as the velocity, V_o, approaches the centerline velocity of such a jet. If this conclusion is valid, then twin jets in their downward travel should first entrain air more rapidly than the single jet and subsequently less rapidly. It would also seem to follow that splitting a jet into small, closely spaced jets should have little effect on entrainment. Experimental verification of these conclusions, or better still, of the manner in which the idealized, twin jets entrain air would further strengthen the authors' thesis.

The authors implicitly assume that the special distribution of momentum flux associated separately with each of 2 jets will not change when the jets are juxtaposed. This assumption is plausible from the point of view of conserving momentum flux, but it may or may not be consistent with the pattern of air entrainment realized in practice.

The principle of superposition as employed by the authors would prove to be a very helpful analytical tool if its limits of applicability were defined and verified experimentally. The supporting experimental evidence presented is encouraging, but should be broadened in scope.

The authors are to be congratulated on having made a contribution to the literature on jets which should have significant implications beyond the immediate objective of their paper.

AUTHORS' CLOSURE (Mr. Austin): I cannot directly answer the first point raised by Professor Helander. It is stated in the paper that we feel there is need for more information. I should not be particularly fearful of using the information in the paper within the limits covered. However, the limits are narrow and, moreover, a great deal of effort is required to obtain data of this kind. We believe that the data have practical value within these limitations and we would like to see the range widened.

There is some doubt about the Reynold's number, and considerable data would be required to arrive at a firm conclusion. It would be interesting to study the effects of greater velocities as well as the use of a wider range of jet diameters.

I doubt if studies at lower velocities would have great value in view of the impracticability of an installation and also because of the fact that it is questionable whether much dependence could be put on data obtained from lower velocities.

Concerning the effect of entrainment as contrasted with superposition, I do not believe we can obtain sufficiently accurate data. Such data are difficult to get, as the least bit of interference from the outside causes trouble. In fact, it is very troublesome to get a non-pulsating jet under any conditions. My feeling, therefore, is what is really needed is more data but that it should be data which is practical and useful for engineering applications.

1581

No. 1581

DEPENDENCE OF WATER VAPOR PERMEABILITY ON TEMPERATURE AND HUMIDITY

By S. C. CHANG*, SASKATOON, SASK., AND N. B. HUTCHEON**, OTTAWA, ONT., CANADA

IN MANY situations a knowledge of the movement of moisture through unsaturated materials is of interest or of importance. These may occur in the design of walls and roofs to avoid condensation, in the design of wetting or drying processes, in the estimation of humidifying and dehumidifying loads, in the selection of packaging to control moisture content, and in the prediction of the moisture content in any material exposed to changing conditions. Many of these cases are of direct concern to the air-conditioning engineer. The need for more and better data is growing rapidly. It is of interest to re-examine the basis on which data are presently obtained and to take stock of the present position.

The movement of moisture through unsaturated material is broadly considered as a vapor flow so long as the water crosses the boundaries of the material or materials in the vapor form. The transmission of water vapor under these conditions is regarded as a vapor diffusion process although under many conditions the water molecules may not all remain in a true vapor state within the material. It is this situation resulting from various interactions between the molecules of water and those of the substance through which it is passing which complicates the study of flow in unsaturated materials and prompts the use of the term moisture rather than vapor. It is with migration of the type normally referred to as water vapor flow with which this paper is concerned.

VAPOR FLOW EQUATION

Engineering calculation of vapor flow has to date been based on a simple flow equation, analogous to that used in heat transfer. The basic form is:

$$w = -\mu(dp/dx) \quad . \quad (1)$$

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** Assistant Director, Division of Building Research. National Research Council of Canada. Member of ASHAE.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956.

where

w = weight of vapor transmitted through a unit area in unit time.
p = vapor pressure.
x = distance along flow path.

and hence:

dp/dx = vapor pressure gradient.

and

μ = coefficient of permeability, or simply, permeability.

This equation is of the same form[1] as the diffusion equation describing Fick's Law which is frequently applied to diffusion processes. The weight of vapor transmitted is proportional to the vapor pressure gradient, provided μ is a true coefficient as implied, and is independent of p and x.

THE PERMEABILITY COEFFICIENT

It would be convenient if the permeability were constant for any particular material. Unfortunately this is not always the case and it becomes necessary to determine if possible the factors which cause it to change and their relation, quantitatively, to it.

Barrer[2] from experimental evidence states that membranes which sorb little water behave at low relative humidity as indicated by Fick's Law, that the more hygroscopic the substance the lower is the humidity above which this relationship breaks down and that the increase in permeability follows roughly the increase in water sorbed.

Joy et al[3] have presented extensive data on sugar pine wood indicating a marked increase in permeability with relative humidity beginning at about 50 percent. They acknowledge the probability that the sorption of water in these materials, in amounts which increase with relative humidity, may be responsible for the increased vapor flow. Babbitt[4] has presented similar data for plywood and 2 types of building paper.

Babbitt[1, 4, 5] assumes that sorption contributes to flow by making possible movement of the water molecules as adsorbed layers on internal surfaces, and believes that the departure of the diffusion from Fick's Law is due to this. Following the method used by Muskat he has analyzed 3 of the currently held adsorption theories and has concluded that the corresponding value of permeability ought to increase with relative humidity at an increasing rate for hygroscopic materials.

At high relative humidities sorbed films become thick enough to merge in small capillaries, filling them. A liquid-type flow then becomes possible. Johansson[6] considered moisture transfer as a combination of capillary movement and gaseous diffusion. The value of permeability which he determined was a function of many factors, including the diffusion constant of air, the size of capillaries, and the viscosity and density of water vapor. He concluded that in general permeability ought to increase with relative humidity although the reverse could be true if the capillaries were specially sized.

Temperature may also have an effect on permeability, leading to different values of permeability at different temperatures for the same relative humidity. Barrer[2] states that changes in temperature have a marked effect on the permeability con-

[1] Exponent numerals refer to References.

stants of the least permeable membranes but that the constants of porous membranes are independent, or slightly dependent upon, temperature. Doty et al[7] found a wide range in the temperature dependence of the permeability of several polymer films of the type used in packaging.

It may be possible to describe the effects of temperature for certain materials on the basis of activation energy. This is a concept developed from kinetic theory to explain the effect of temperature upon rate of chemical reactions. It is based upon the premise that only the molecules having kinetic energies above a certain level

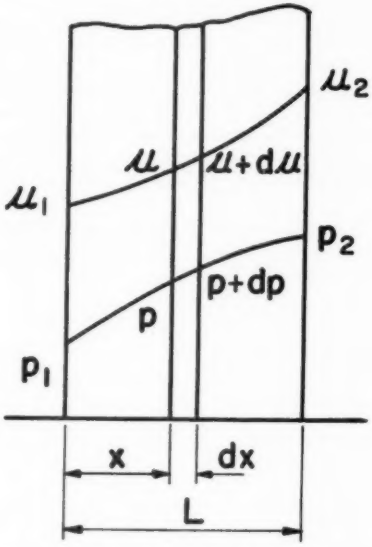

FIG. 1—VARIATION OF VAPOR PRESSURE AND PERMEABILITY THROUGHOUT A HOMOGENEOUS SAMPLE

can enter into a reaction. Its application to certain types of diffusion processes[2] leads to an equation of the form:

$$\mu = \mu_0 e^{-E/RT} \quad . \quad . \quad .

Equation 2 is readily applied to data in a graphical solution when transformed to:

$$\log \mu = \log \mu_0 - (E/RT) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

A straight line of slope $-E/R$ is obtained in a plot of $\log \mu$ against $1/T$. The intercept with the Y-axis is $\log \mu_0$. Any 2 values of μ for different temperatures but at the same relative humidity may be used to establish the slope of the line, and, therefore, the value of E, from which the coefficients for other temperatures may be established.

THE AVERAGE PERMEABILITY

When moisture is transmitted through a material the temperature and humidity may vary along the path of transmission. As the discussion of the previous section indicated, the permeability, μ, is likely to be, in practice, a function of relative humidity and temperature and to have different values from point to point.

Some simplification may be introduced if the possibility of a temperature gradient is disregarded, and only isothermal cases considered. The permeability μ can then be regarded as a function only of vapor pressure (or relative humidity) at any particular temperature.

The conditions for uni-directional, steady-state flow through a slab of material of thickness, L, can be represented as in Fig. 1, and Equation 1 applied.

Equation 1 may be written:

$$w dx = -\mu \, dp \quad . \quad (4)$$

Integrating Equation 4 from $x = 0$ to $x = L$, and from p_1 to p_2 and dividing both sides by the thickness:

$$w = \frac{-\int_{p_1}^{p_2} \mu \, dp}{L} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

or

$$w = \left[\frac{-\int_{p_1}^{p_2} \mu \, dp}{p_2 - p_1} \right] \left[\frac{p_2 - p_1}{L} \right] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

transposing

$$\frac{wL}{p_2 - p_1} = \frac{\int_{p_1}^{p_2} \mu \, dp}{p_1 - p_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

let

$$\frac{\int_{p_1}^{p_2} \mu \, dp}{p_1 - p_2} = \bar{\mu} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

Then Equation 6, rearranged, becomes:

$$w = \bar{\mu}(p_2 - p_1)/L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Equation 9 is in the form most commonly used in calculations of vapor flow. The coefficient $\bar{\mu}$ is usually found by the application of this equation to the results

of dry-cup, or wet-cup tests carried out on sheets or plates of material at one particular temperature. The results are frequently expressed as a permeance, $\bar{\mu}/L$, which applies to the thickness of material stated or implied, rather than to unit thickness. The values obtained may, however, be unique to the conditions of test since $\bar{\mu}$ is the average coefficient of permeability over the range of vapor pressure in the particular test.

The relationship between μ and $\bar{\mu}$ can be more readily seen by reference to Fig. 2 where the permeability μ is plotted against vapor pressure for the conditions represented in Fig. 1. The coefficient $\bar{\mu}$ is numerically equal to the area ABCE divided by AB, as given by the vertical ordinate to the mean line GH. It may be noted that area GFE in Fig. 2 equals the area FCH and that generally the average per-

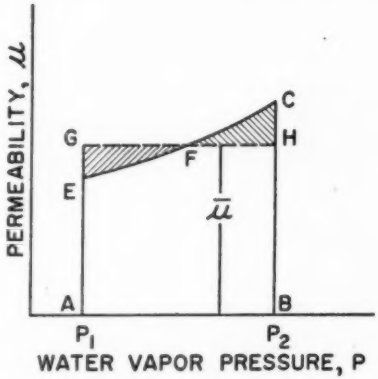

FIG. 2—VARIATION OF PERMEABILITY WITH VAPOR PRESSURE, AT CONSTANT TEMPERATURE

meability $\bar{\mu}$ does not equal the *spot* or *differential* permeability μ at mean vapor pressure $(p_1 + p_2)/2$.

A further representation of the average permeabilities obtained from dry-cup and wet-cup tests carried out at the same temperature but at relative humidities of 0 percent to 50 percent and 50 percent to 100 percent respectively is given in Fig. 3. In this case the plot is against relative humidity, since at constant temperature vapor pressure and relative humidity are for practical purposes proportional. For a curve of permeability varying with relative humidity as shown, the average permeabilities obtained will be $\bar{\mu}_1$, for the dry-cup test and $\bar{\mu}_2$, for the wet-cup test. These frequently differ by a factor of 3 or more for wood and wood products, the wet-cup test providing the higher values.

THE PERMEABILITY CURVE

The permeability μ cannot be measured directly. It is possible, however, from a series of values of average permeabilities for a particular material at a given tem-

perature obtained from suitable tests, to construct a permeability curve such as that shown in Figs. 2 and 3. If in Fig. 3 the average permeabilities for the dry-cup and wet-cup tests were plotted as shown it would be possible to construct a rough permeability curve on a trial and error basis.

The trial curve must satisfy the condition in the case of each of the tests that the area under the curve so constructed between the limits of relative humidity for the test should equal the area under the average permeability line as found from the tests, between the same limits. Consideration shows that 4 dry-cup tests run at humidities outside the cup increasing in increments of 20 percent would permit a reasonably accurate curve to be drawn, up to the high humidity range.

Once the permeability curve is obtained it is possible to determine by a reverse

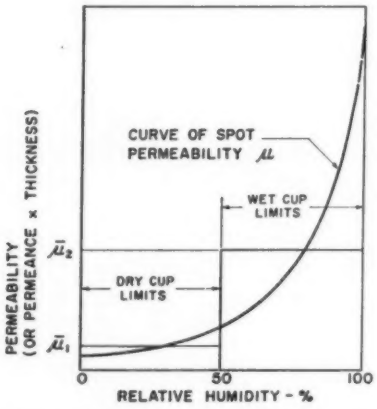

FIG. 3—RELATIONSHIP BETWEEN DRY- AND WET-CUP TESTS AND THE SPOT PERMEABILITY FOR A MATERIAL SUCH AS WOOD

procedure the average permeability $\bar{\mu}$ for any desired limits of humidity (or vapor pressure) for isothermal conditions at the temperature for which the test data were obtained.

A more straightforward method suggested by a colleague involves plotting the values of the integral $\int_{p_1}^{p_2} \mu \, dp$ calculated from the dry-cup test data rather than

$$\frac{\int_{p_1}^{p_2} \mu \, dp}{p_1 - p_2}$$

which is $\bar{\mu}$.

Babbitt[4] used the device of plotting permeabilities, or permeances where appropriate, against the mean relative humidity for the test conditions, and of draw-

ing a curve through the points obtained from a series of tests. This curve can provide data for other mean relative humidities, and is in effect, an approximate permeability curve. The nature of the approximation is evident from Fig. 3, *i.e.*: the true average permeability is not the permeability at the mean humidity. This method may, however, be sufficiently accurate for many purposes.

Further curves of permeability applying to other temperatures can be constructed from test data, for such other temperatures. Some reduction in the amount of

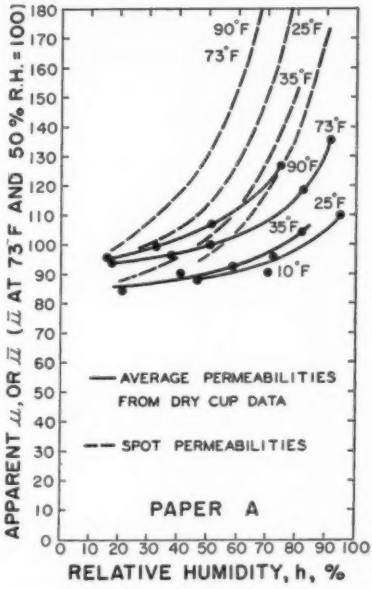

FIG. 4—PERMEABILITIES OBTAINED FROM DRY-CUP TESTS OF A 15 LB ASPHALT-SATURATED SHEATHING FELT, 0.037-IN. THICK

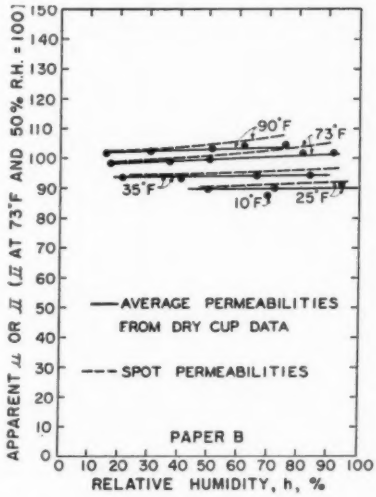

FIG. 5—PERMEABILITIES OBTAINED FROM DRY-CUP TESTS OF A LIGHTLY SATURATED AND WAXED KRAFT BUILDING PAPER, 0.008-IN. THICK

testing required to cover a range of temperatures may be obtained by using Equation 2 to extend the data obtained for any two temperatures to other temperatures.

PAPER TEST DATA

The variation of permeance with temperature and humidity was determined for 3 building papers by an extended series of dry-cup tests. The results obtained for 2 of these, an asphalt saturated felt, A, and a smooth, lightly saturated and waxed kraft building paper, B, are given in Figs. 4 and 5, respectively. They are ex-

pressed as relative permeabilities, as percentages of the test permeability found at the standard *Testing Association of Pulp and Paper Industry* test conditions of 73 F and 50 percent relative humidity. The solid curves are the average permeabilities found from the tests, plotted against the relative humidity outside the cup (0 percent on the inside). These data were reduced to apparent permeabilities, by the method outlined in this paper, which are represented by the dotted curves.

The very marked influence of temperature and humidity on permeability for paper A of Fig. 4 may be noted. Of interest is the reduction in slope of the permeability curve at the higher relative humidities as the temperature is reduced. The error introduced by applying data from the standard dry-cup test conditions to higher humidities without correction and the increasing effects of temperature at higher humidities are apparent. The third paper, an asphalt-saturated building

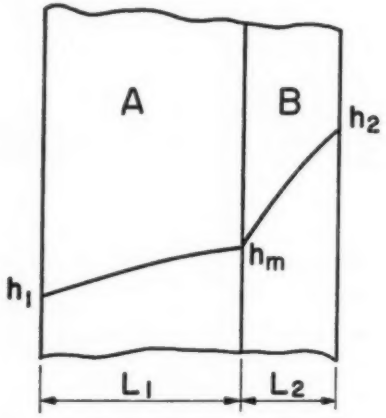

FIG. 6—SIMPLE CASE OF A COMBINATION OF MATERIALS

paper showed similar results. Paper B of Fig. 5 gave markedly different results, showing only small increases in permeability with increase in temperature and relative humidity. Hygroscopic moisture data have not yet been obtained for these papers, but similar results might be expected since both are essentially wood products. These results are interesting in view of the relationship which has been proposed between hygroscopicity and increase in permeability with relative humidity, and suggest that another factor must be taken into account. Paper B has a relatively low expansion with increase in moisture content.

PERMEABILITY OF A COMBINATION OF HOMOGENEOUS MATERIALS

The average permeability of a combination of homogeneous materials can be determined if the appropriate permeability data are available for each material. This is a procedure of much interest in practice, particularly in calculations of vapor transmission through walls and roofs. The resistance method of calculating

an overall coefficient which is employed in the case of heat transmission, by adding the resistances of the various elements, can be used. It must be noted, however, that the individual resistances and, therefore, the overall coefficient, in the case of vapor flow, may vary with humidity in the isothermal case, with both temperature and humidity, and perhaps even with moisture content gradient, when there is a temperature gradient. The humidity limits for each element must be assumed before a permeance value can be selected for it, thus leading to a trial and error procedure.

Another method based on permeability curves can be used for the isothermal case. Fig. 6 illustrates a combination of 2 materials A and B of thickness L_1 and L_2, respectively. Material A is assumed to be highly permeable and B less per-

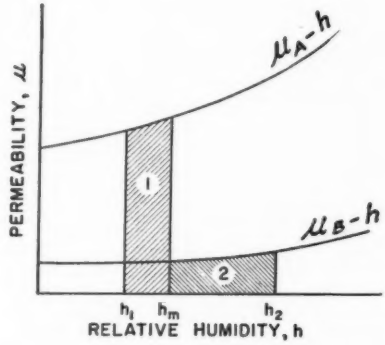

FIG. 7—PERMEABILITIES AND RELATIVE HUMIDITIES FOR A SIMPLE COMBINATION OF MATERIALS

meable, with their corresponding μ-h curves shown in Fig. 7. The ambient humidities are h_1 and h_2 and the interfacial humidity is h_m. The problem is essentially one of finding the interfacial humidity h_m for which the rate of flow will be the same through each of the two materials. It follows from Equations 8 and 9 that the area under the permeability curve between appropriate limits of relative humidity gives the flow rate through unit area for unit thickness of material. The flow rate for thickness, L, is therefore given by the area divided by L. The interfacial humidity, h_m, must therefore be selected so that area 1 of Fig. 7 is to area 2 as L_2 is to L_1. Curves of permeance, or μ/L, might be used, in which case the corresponding areas 1 and 2 would be equal, for equal flow rates. Once the appropriate interfacial humidity is known, the average permeability and the average permeance of the combination can be calculated.

A reversal of the humidity conditions with A exposed to humidity h_2, and B exposed to humidity h_1, will normally change the value of h_m and the overall average permeability as shown in Fig. 8. The change in $\bar{\mu}$ resulting will depend on the relative thickness of the two materials and on the shape of their respective μ-h

curves. This effect is common in the dry-cup testing of coated papers where different results are obtained with different orientation of the test specimen.

The method outlined may be extended to combinations of more than two materials. It does not, however, avoid the need for trial and error in determining the interfacial humidity. As in the case of the construction of the permeability curve, this operation can be facilitated by employing plots of the integrals of the permeability curves so that the new curves give the areas under the original permeability curves at successively higher humidities directly. The need for repeated measurement of areas for comparison is thus avoided since these can be determined

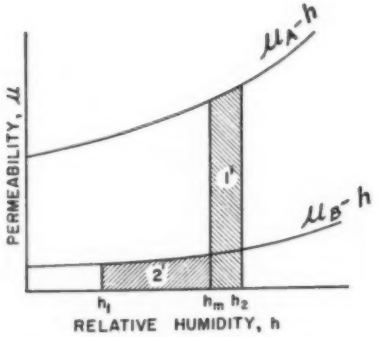

FIG. 8—PERMEABILITIES AND RELATIVE HUMIDITIES FOR A SIMPLE COMBINATION OF MATERIALS AS IN FIG. 7, BUT FOR THE MATERIALS IN REVERSE ORDER

for particular relative humidity limits from differences in vertical ordinates to the integral curve.

EFFECT OF THICKNESS

The method presented for determination and use of permeability curves is predicated upon an assumption not yet stated explicitly, that the permeability coefficient is independent of vapor pressure gradient. Stated in other words the permeability is assumed to be independent of the thickness. Babbitt[1, 5] has presented experimental evidence in support of this as have Joy et al[3].

There is some evidence[2] indicating that in certain cases thickness may be a factor. If so, this introduces still another complication in the case of homogeneous materials which may be tested and used in a range of thicknesses, since a simple correction for thickness cannot then be made. It may well be that such effects are to be found at high humidities, particularly for thin films or membranes.

Materials which are non-homogeneous as to moisture properties ought always to be dealt with on the basis of permeance, rather than on a unit thickness basis, and should be tested in the thicknesses used. Results may, however, be expressed

as average permeabilities so long as it is recognized that these apply to certain specific thicknesses or arrangements.

EFFECT OF A TEMPERATURE GRADIENT

Experimental determination of vapor flow through a material under the combined effects of both vapor pressure and temperature gradients is exacting and time-consuming. Few such tests have been conducted, and are not particularly revealing as to the influence of a temperature gradient. In the combined gradient case, the relative humidity is no longer proportional to vapor pressure. It is possible, depending on the temperature gradient, to have the relative humidity increasing, decreasing, or increasing and then decreasing along the flow path through the material.

If permeability is a function only of relative humidity and temperature, the methods presented for the isothermal case can be elaborated to deal with the non-isothermal cases. Babbitt, however, attributes the increase in permeability with humidity to the increased sorption and to the migration in sorbed layers under a gradient in sorbed moisture. In the isothermal cases the vapor pressure, relative humidity and sorbed moisture concentration all decrease in the direction of flow. However, with certain temperature gradient conditions, the relative humidity and, therefore, the sorbed moisture concentration can be made to increase in the direction of decreasing vapor pressure[8]. It is thus possible that a vapor diffusion may take place in the direction of decreasing vapor pressure, and be opposed by a film, or by a combined film and capillary migration occurring in the opposite direction.

Edenholm[9] has proposed that moisture migration be expressed in terms of two mechanisms, one a *diffusion* depending only on vapor pressure, and the other a capillary movement depending on a moisture content gradient. He uses separate coefficients for each mechanism, but combines them in one equation. He has further proposed a method by which the coefficients may be obtained and combined for use in a graphical plot. While his method contains many approximations, it is interesting for it attempts to deal with combined vapor pressure and temperature gradients.

Further experimental investigation of such cases is greatly needed.

CALCULATION OF FLOW UNDER COMBINED GRADIENTS

Calculation of vapor flow under combined temperature and humidity gradients can presently be carried out, using the resistance method, based on the simple flow equation. This is a rational method, so long as the permeabilities are reasonably independent of both temperature and humidity. Briefly, this means that it will best apply to the low-to-moderate range of relative humidities for porous, non-hygroscopic materials.

There are at present very limited data from which a selection of permeabilities for design may be made. These are mainly from dry-cup and wet-cup tests and therefore are strictly applicable only to isothermal flow cases. Judgment must be used in selecting design values. It may well be that the simple flow equation based only on vapor pressure as the potential causing flow may have to be replaced by one based on a more involved concept of the mechanism or mechanisms of migration. Until then, it would seem to be best to rely on the simple flow equation, and to use permeability data related as well as possible to the relative humidities

in use. The complications of film or capillary migration which may oppose the main vapor flow must be left for further investigation.

CONCLUSION

Complications in the determination of permeabilities or permeances which have been discussed arise from the use of a simple flow equation. Much of the complexity of the actual functional relationship between flow rate, material properties, and specimen environment, is thus thrown into the permeability coefficient. Whether it is possible to develop a new flow equation with constant coefficients to describe the material properties remains to be seen. The simple flow equation can in the meantime be used, although with difficulty, if the permeability coefficients can be evaluated. It will be a forward step if the nature of these coefficients can be generally recognized, as an aid in stimulating investigation, and promoting rational procedures.

The adequacy of a single test of a material at one set of conditions ought not to be taken for granted. Tests ought to be carried out at two different humidity conditions and at two different temperatures to indicate the degree of dependence of permeability upon these factors. Whenever possible a series of tests over a range of conditions should be carried out for each material. This is the only way at present by which the permeability of a material can be properly evaluated.

Dry-cup tests are relatively simple, although time-consuming. Data from them can be used, by the methods outlined, to establish a permeability curve applicable to the isothermal case. Permeability data over a range of temperature and humidity are available for only a few materials. A very great deal of work will be required to cover even the more common materials. Variations in properties from one sample to another are frequently great and lead to an increase in the number of samples to be tested.

Research must be carried out to establish more definitely the nature of the dependence of permeability upon temperature, relative humidity, and thickness. It may then be possible to find ways of reducing the amount of testing required for evaluation of a particular material.

The discussions presented have dealt largely with flow under isothermal conditions. There is great need for some method of dealing with the general case of a temperature gradient together with a vapor pressure gradient. This will provide more complications than the isothermal case, since under a temperature gradient relative humidity and vapor pressure are no longer directly proportional, and their relationship is different for each new set of temperature conditions. The gradient of concentration of sorbed moisture which is largely dependent upon relative humidity may differ in relation to vapor pressure and to the vapor pressure gradient from one case to the next. As a result, the migration of sorbed moisture may for certain conditions oppose the vapor flow, whereas in the isothermal case these are more simply related and are always additive.

ACKNOWLEDGMENTS

This paper is a contribution from the program of cold weather wall research being carried out co-operatively between the University of Saskatchewan and the Division of Building Research, National Research Council. The paper test data presented were obtained from a post-graduate research project carried out by the senior author. The authors are indebted to G. O. Handegord, officer-in-charge of

the Prairie Regional Station of the Division, for his interest and assistance, and to Dr. T. D. Northwood of the Building Physics Section of the Division, at Ottawa, for the suggestion to use the integrated permeability curve.

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DISCUSSION

W. A. DANIELSON, Memphis, Tenn.: Two years ago, I air conditioned my house. It is largely paneled in wood frame throughout, no vapor barriers. The 45 deg cooling water condensed about a half to one gallon of water an hour, and continued to do so. I think this question of vapor barrier is so important that each one of us should stress the necessity of including this in the original building design. It not only applies to residences but applies to office buildings too. In a large office building constructed in Memphis—the main office being paneled of plywood—the moisture came through back of the plywood, damaged it so extensively that court action almost resulted.

Each one of us should spread information about the importance of providing vapor barriers—particularly to architects.

R. C. LAST, Melbourne, Australia: I would like to ask Mr. Hutcheon if he has any actual figures for common building materials so that we, as air conditioning engineers, can have some knowledge of the physical values of moisture transmission.

Another question—Could he outline for those present the dry cup test? I am afraid I haven't read the paper and I would appreciate a discussion on this.

AUTHORS' CLOSURE (Doctor Hutcheon): There are many vapor permeance data available on many materials. Those of us who are concerned with winning such data, however, have long been conscious of the great variations in the values obtained with differences in testing conditions. Presumably, there are also differences in the actual values which might be used in design, depending not only upon the variations in the conditions under which the vapor flow takes place, but also upon the variations in materials and constructions.

It is because of this problem which has been with us for many years that we have been unable to turn out all kinds of data quickly. If it were possible to set up single values for each kind of material, this would have been done long ago. It is quite clear that one must first understand the nature of the variations one is trying to deal with and the reasons for them before he can hope to turn out good data in any reasonably expeditious way.

With regard to the dry cup test, it is regretted that more time was not devoted to a description of it. It is a relatively simple test for measurement of vapor permeability, involving the use of a shallow cup perhaps 3 or 4 inches in diameter containing a suitable desiccant. The sample to be tested, which is usually a thin membrane such as a building paper or a plastic film, is sealed with wax to the top of the cup. The cup is then exposed to a constant temperature condition at a known humidity. Water vapor flows under the difference in vapor pressure through the sample into the cup where it is taken up by the desiccant. The permeability or the permeance can be calculated for the conditions of the test from the rate of moisture gain which is determined by periodic weighing. The dry cup test is usually carried out at conditions of 73 F and 50 percent relative humidity.

The wet cup test is not so commonly used. In it, water is placed inside the cup, providing 100 percent humidity, and a lower relative humidity, usually 50 percent, is maintained outside.

1582

No. 1582

EFFECT OF UNBALANCED AIR PRESSURE ON PERMEANCE

By F. A. Joy* and D. R. Fairbanks**, University Park, Pa.

MOISTURE is vital to all life, but its behavior is both blessing and bane in modern living. The blessings we accept. Until recent years the damages of moisture were also accepted with only a mild protest. Increasingly, however, attention is directed to the prevention of these damages—the corrosion of metals, the rotting of wood, the blistering of paint on houses—to cite only a few. Today a huge industry provides packages that retain moisture in lettuce or exclude it from crackers. Buildings also are packages.

A vapor barrier is required in most buildings and its effectiveness in reducing exterior paint damage and moulding within frame structures has been demonstrated. This damage is a winter phenomenon that has developed, not because paint is poorer or wood is different from what it was in grandfather's day, but rather because housing standards and habits have changed. Houses are built smaller and tighter. More water is used for bathing and for automatic kitchen and laundry devices that frequently discharge steam into the house. With no fireplace to draw this water vapor up the chimney, it condenses on the windows and in hidden places within the structure unless a vapor barrier prevents its entrance thereto. Without such moisture control, the house walls and roof may become wet in winter. This moisture migration is outward.

When air conditioning is used, moisture tends to move into the building in summer. This movement rarely damages the building but it adds to the cooling cost and may even overload the cooling equipment and render it ineffective.

Residential or commercial buildings cooled only for comfort furnish no major difficulty in moisture control. Much more difficult are the problems in the construction of cold rooms, refrigerated vessels and pipe lines. In these cases, moisture enters from the exterior and accumulates in the insulation increasing its heat transmission and sometimes seriously damaging its structure. As the cold space may never be warm, there is no season for drying out the insulation, and the moisture

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** Engineer, Carrier Corp.; formerly Research Assistant at The Pennsylvania State University.

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accumulation mounts from year to year. It is evident that a vapor barrier for such applications should be highly efficient. A considerable cost can be justified.

MECHANISMS OF MOISTURE MOVEMENT

The causes of moisture migration have been discussed by many authorities[1, 2, 3] and the following mechanisms have been recognized:

1. Molecular diffusion through air in a porous body.
2. Molecular diffusion through a nonporous solid.
3. Film spreading (adsorption and absorption) within a porous body.
4. Transport in a mass movement of air.
5. Capillarity.
6. Osmosis.
9. Gravity.

Many observers have placed emphasis on the first mechanism and undoubtedly it is the most important in many situations. For this case, Fick's Law for vapor diffusion applies. For a porous sheet material, the relation is:

$$W = Mat(\Delta p) \quad \ldots \quad (1)$$

where

W = Weight of the vapor transmitted through area a in time t when the vapor pressure difference across the sheet is Δp.
M = permeance of the sheet as it is made and used.

This factor has often been called its *permeability*, but such usage is improper since *permeability* is a property of the substance, numerically equal to the permeance of a unit thickness. Fortunately, these terms are now being used correctly. They have been standardized by the *ASTM* and there is much less confusion. The unit of permeance used in the building industry is a *perm*, defined by *ASTM* as one grain per (sq ft) (hr) (in. Hg vapor pressure difference).

But Fick's Law does not apply to any of the other 6 mechanisms and 1 or several of these may be operating at the same time as vapor diffusion through pores. Most complicating is the fact that when 2 mechanisms are operative they are not independent of each other. A relation that defines the combined result of all mechanisms would be complicated and it has not been formulated. Instead, Equation 1 is customarily used. This is generally adequate for engineering work when it is recognized that the coefficient M in the equation is not a constant for all conditions. The value of M (or permeance) for a given sheet must be selected to fit the exposure conditions. With complete information about its value under all conditions, the choice is properly made and the computation is correct in spite of the complicated behavior within the sheet that is not fully understood. Unfortunately, however, there is not complete information of this kind for many materials that are, or may be, important for engineering use. Indeed, the information is so scanty that only the basic concepts of water vapor transfer are readily available.

FACTORS AFFECTING PERMEANCE

In response to this need, an investigation was undertaken in 1952 in 3 areas as follows:

1. Effect of unequal total pressures on water vapor permeance.
2. Effect of low temperature on water vapor permeance.
3. Effect of relative humidity on water vapor permeance.

[1] Exponent numerals refer to References.

This paper confines itself to the first of these 3 areas.

Unbalanced pressure on a vapor-resisting sheet or structure is not an unusual condition in practice. The pressure of wind on a house is likely to be felt by the vapor barrier installed in its walls. In cold-room construction, the greater density of air within the room produces an unbalance outward near the floor; inward near the ceiling (modified if there are vents). In its walls, a potential unbalance exists since the air therein has a different density from the air on either side. These facts raise a question whether the standard cup tests are adequate to determine the performance of vapor barriers and other members in such service.

The work here reported was done on a limited number of sheet materials, chosen primarily for their diverse characters rather than for their importance as vapor control membranes in buildings. This diversity provides a broad view of the effect of total pressure, leading to generalized conclusions.

APPARATUS AND METHODS

The test cell, sometimes called the *Banjo Cell* was the basic apparatus. This has been described in an earlier paper[4]. The *Model 5* cell shown in Fig. 1 has 2 chambers between which the test specimen is installed in a 12-in. diameter ring. In one chamber, a water container hangs on a calibrated spring balance. In the other, a desiccant container is similarly supported. In each chamber, a fan circulates the air and assists in the maintenance of the chosen relative humidity. These fans are driven through magnetic couplings by external motors. Control of relative humidity is accomplished by an electric hygrometer and associated electronic equipment. For these tests, the cells were placed in a temperature controlled room, providing isothermal test conditions. These arrangements insure a well defined vapor pressure difference across the specimen. The rate of moisture transfer is obtained by periodic readings of the spring extensions. When the rate of loss from the water source matches the rate of gain in the desiccant sink, a steady state exists, and test results are valid. Only steady-state conditions are here considered.

The schedule for each specimen required first a test of its permeance with no unbalance of total pressure across it. Subsequently, tests were made with a *positive* unbalance (in the direction of vapor pressure difference) and then with a *negative* unbalance. The total unbalance ranged up to 4 in. water pressure and required a wire grid support for some specimens. In all tests, it is convenient to relate the vapor transfer rate to the vapor pressure difference using Equation 1. For distinction, the value of M with unbalanced total pressure is called *apparent permeance*.

To provide this unbalance of total pressures, 2 methods were employed, the unbalance being measured by an inclined differential manometer.

1. The *closed* method was used when the specimen was not porous to air. It required only a small charge of excess air in 1 cell chamber or the other at the start of the test.

2. In the *circulating* method, air was supplied at low pressure to 1 chamber and discharged out of the other to waste. However, the supply air was thoroughly dried before entrance and the exit air was also dried before wasting. In the former case, no water was added to the system. In the latter case, the drier is considered a part of the system and its water pick-up included in the calculation. For dependable and equal drying of supply and discharge, 2 or more driers on each side were connected in series. Each contained a desiccant and any appreciable gain in the last of the series was a signal for changing the desiccant in all.

The air porosity of each specimen was measured before and after each permeance test in a separate apparatus using a water displacement technique. The second measurement was required to detect any damage to the seal of the specimen in its mounting that might have occurred during the test. Within the limited range of

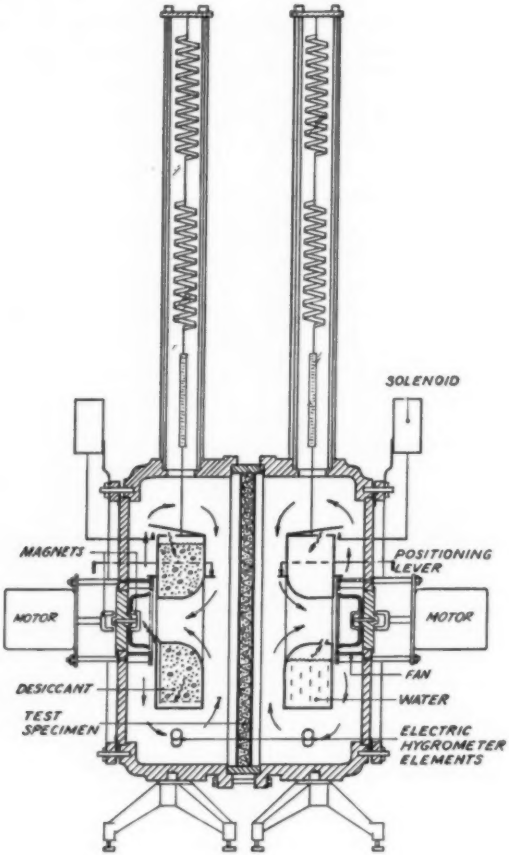

FIG. 1—MODEL 5 BANJO CELL

air pressure utilized, the air flow rate for each specimen may be considered proportional to the pressure difference, the factor being its *porosity coefficient*, expressed in Table 1 as grains of dry air per (square foot) (hour) (inches H_2O).

RESULTS

Five materials were tested as shown on Table 1, but only the results from the first 4 are reported in full. The air porosity of 2 was nil or slight and of the other 2 was relatively high. In all, 46 separate tests were made. The results of 29 are shown in Fig. 2 with the apparent permeance plotted against total pressure difference. It is immediately evident that the permeance of cellulose acetate was unchanged in the range of pressure applied, while the apparent permeance of the porous 15 lb roofing felt increased as much as 25 fold when the total pressure dif-

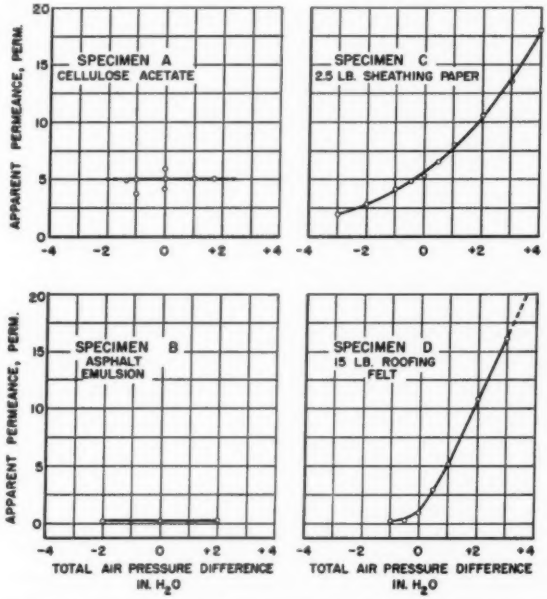

FIG. 2—APPARENT PERMEANCE OF FOUR SHEET MATERIALS

ference was positive. Naturally, the effect of negative unbalance was less conspicuous.

The results indicate that the permeance is affected by air flow and not by air pressure, *per se*, in the test range. It also appears that the effect of air flow is sometimes so large that the simple coefficient, *apparent permeance*, is not a desirable way to express this effect.

For the porous sheets, the air flow may be considered largely a mass movement and the moisture transfer treated as consisting of 2 parts: (1) the normal vapor diffusion; (2) the transport of vapor by mass air flow as expressed in Equation 2, thus:

$$W' = M_0 \Delta p + Fh' \quad . \quad (2)$$

where

W' = total vapor transfer in grains per (square foot) (hour).
M_0 = normal permeance (with balanced total pressures).
F = air flow rate (dry air) in grains per (square foot) (hour).
h' = *effective moisture content* of the air flow in grains per grain of dry air.

Letting h = the actual moisture content of the air on the side of higher vapor pressure and putting $h'/h = E$, we have:

$$E = (W' - M_0\Delta p)/Fh \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

In this form, E is a ratio expressing the efficiency of the air flow as a carrier of vapor. Naturally, $E = 1$ for positive air flow through an orifice.

TABLE 1—SPECIMEN MATERIALS AND TEST CONDITIONS

SPECIMEN AND DESCRIPTION	AIR POROSITY GRAINS DRY AIR PER (SQ FT) (HR) (IN. H_2O)	TEST CONDITIONS			PERMEANCE WITH BALANCED PRESS. PERMS	TEST METHOD
		TEMP. F	PERCENT RH			
			SOURCE	SINK		
A Cellulose Acetate, 10 mil sheet	0.00	70	62	2	5.1	closed
B Asphalt Emulsion[a] ⅛ in. coat on ¾ in. cork board	0.27	70	90	1	0.20	closed
C 2.5 lb Sheathing Paper, asphalt infused kraft	157.	72	75	5	5.5	circulating
D 15 lb Roofing Felt, asphalt saturated, 15 lb per 100 sq ft	220.	72	75	5	0.82	circulating
E Cement Asbestos, ¼ in. board	0.14	70	74	1	2.0	circulating

[a] Sand filled clay type asphalt emulsion, trowelable grade.

In any test, all factors on the right side of Equation 3 are known and E can be found. This has been done for Specimens B, C and D. (Specimen A has zero air flow and E is indeterminate). The ratio is plotted in Fig. 3.

The significance of this ratio, E, is evident when the air flow is positive (same direction as vapor pressure). For negative air flow, E is still positive but its physical meaning is less clear. However, by relating all test results to the same base, their correlation is continuous through zero air flow, facilitating analysis.

In Fig. 3 for positive flow, E ranges up to 1.0 as a limit, indicating that the efficiency of air as a vapor transport is lower in a porous medium than in a simple orifice. This appears to be due to loss of freight from the carrier along the way. It indicates that the 2 mechanisms—diffusion and transport—are not independent, confirming the principle stated earlier.

On the negative side, the same inter-relation is seen. The counter-flowing air, starting essentially dry, picks up moisture along the way until it emerges with a substantial load—larger per unit of air when its flow is small. It will be noted that the tests did not show a net vapor transfer against the vapor pressure gradient.

Of course, this would be unlikely with the low-side vapor pressure that existed. With higher vapor pressure on the low-side, negative air flow would have a higher efficiency and readily transport more vapor than could diffuse against it, thereby providing a net transfer counter to the vapor pressure gradient.

Comparing one material with another plotted on the same scale in Fig. 3, marked differences are seen. For Specimen C, 2.5 lb sheathing paper, the efficiency increases slowly and smoothly throughout the range of air flow, suggesting an intimate inter-relation with the diffusion mechanism. This material is believed to

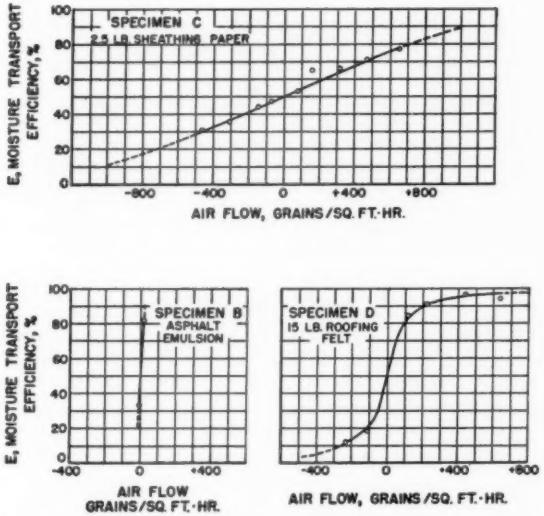

FIG. 3—EFFICIENCY OF MOISTURE TRANSPORT BY AIR FLOW THROUGH POROUS SHEETS

have very small but innumerable air paths. In contrast, for Specimen B, ⅛ in. asphalt emulsion on corkboard, the transition from negative to positive air flow is sharp. This result suggests that the air paths are few and essentially orifices; possibly defects in the sheet. There is little chance for inter-connection with the vapor diffusion mechanism. In Specimen D, 15 lb roofing felt, the effect is intermediate; suggesting a coarse structure open to air flow.

These results are significant in practical vapor control problems. Where a vapor barrier membrane is placed in a wall cavity, air movement around the margin of the membrane or through joints or tears in it, is a potent means of vapor transport. Such air movement entirely within a wall cavity invariably has the net effect of carrying moisture to the zone of lower vapor pressure and possible condensation.

On the other hand, air that moves into a wall structure from a drier (colder) source may be expected to remove moisture from the structure when it leaves.

Practical application of this principle is found in house walls properly vented to the weather. Similarly, attempts have been made to apply this principle to cold room walls vented to the cold space.

In any structure, it appears possible to develop slight pressure by introducing adequately dry air into the cold side of the assembly and thereby reduce vapor entrance through a vapor barrier and especially through its defects. Of course, the economics of such applications involve the altered heat transfer through the structure and, in the case of a cold room, the need for cooling the air supply.

CONCLUSIONS

1. The diffusion of water vapor through a membrane is not affected by a small unbalance of the total air pressure across it when no air transfer results.
2. When air passes through a membrane because of wind or other small unbalance of pressure it carries moisture with it. When the air flow is in the same direction as the vapor pressure gradient it increases the moisture transfer. When opposite, it reduces the moisture transfer and may reverse it.
3. Air, filtering through a porous membrane in the same direction as the vapor pressure gradient delivers less moisture than the same air quantity blowing through a hole or crevice; but air flow opposite to the vapor pressure gradient delivers more moisture through a porous membrane than it could through a hole or crevice.
4. Air flow, through or around, a vapor barrier in the same direction as vapor pressure can seriously reduce its effectiveness.

ACKNOWLEDGMENT

This report and 2 others planned for early publication are based on work done in the Thermal Research Laboratory at The Pennsylvania State University under the sponsorship of the Armstrong Cork Co. F. M. Gavan and R. J. Calvert, representing the sponsor, have contributed much to its accomplishment.

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DISCUSSION

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in its installation suffers proportionately. The investigation described in the paper promises a means by which the effect of variations from normal practices can be evaluated.

However, examination of the results of the tests on the limited number of materials, indicates that the one material, asphalt emulsion, which meets the FHA requirement of an adequate vapor barrier was not affected by difference in the total pressure. Its apparent permeance remained constant from -2 in. to $+2$ in. H_2O. It would therefore be enlightening to have data on a number of materials whose permeance, tested by present closed cup methods, is not more than unity.

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I believe we have a much more important job than we are doing and I hope that from some of these papers we can get some literature that is factual and definite and which we can build on.

Now, you speak, Professor Joy, on the subject of materials. What is the answer to the *air curtain* question? We are forced to answer this now as one of the problems of air conditioning.

In summer air conditioning we want to keep the door open and from there, we proceed to the whole open wall. We must have *air curtains.*

What is the moisture migration in the rain, on a stormy day at 95 F against this inside comfort condition? We have no facts.

I repeat the work in the field is solving the problem, right or wrong.

AUTHORS' CLOSURE (Professor Joy): On the subject of absorption of moisture in a vapor barrier, it appears to me that with our test apparatus, the determination of moisture transfer was independent of the amount of water absorbed in the barrier. We measured the transfer after a steady state existed, after we had an absorption in the barrier which placed it in equilibrium with the exposure medium. So I do not think this was a factor in our results.

Mr. Avery has stated correctly that our research is not keeping up with the needs as well as it should, and we are plugging away at it.

Vapor transfer is a slow sort of investigation. Just recently, we recorded the results on tests that had been running 3 years. Ordina.ily, we can get a cup test done in a month, but with this procedure, cup testing is necessarily rather slow. Moreover, the research isn't glamorous enough to attract some of the assistants that we would like.

1583

No. 1583

LABORATORY FOR SOLAR ENERGY STUDY AT MINNESOTA

By R. C. JORDAN* AND J. L. THRELKELD**, MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, in cooperation with the Graduate school, University of Minnesota, and conducted through the Department of Mechanical Engineering, University of Minnesota, Minneapolis, Minnesota.

SOLAR radiation is our single large source of inexhaustible energy. An accelerated interest in solar energy has developed in recent years and a number of symposia have been held in various parts of the world. Several institutions in the United States are now engaged in sizeable research projects on solar energy. One of these is at the University of Minnesota.

Of the many problems involved in the harnessing of solar energy, 2 of the more important ones are (1) the extent and availability of the energy supply, and (2) the design and performance of flat-plate collectors. To contribute further knowledge on these 2 topics has been the primary reason for establishing the solar engineering laboratory described here.

The availability of solar radiation outside the atmosphere is accurately known. But in passing through the atmosphere part of the radiation is scattered and part is absorbed by the constituents of the atmosphere. This depletion is substantial even during cloudless days, and with heavy cloudiness depletion of direct solar radiation may be almost complete.

Solar radiation received by a surface on the earth may consist of direct radiation from the sun and indirect or diffuse radiation from the sky vault and from objects around the irradiated surface, particularly the ground cover in front of the surface. In order to design a solar utilization device, detailed knowledge of rates of direct and diffuse radiation are necessary.

Important contributions have already been made toward accurately estimating cloudless day radiation at the earth's surface. Notable are Moon's paper[1] on

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** Associate Professor, Department of Mechanical Engineering, University of Minnesota. Member of ASHAE.

[1] Exponent numerals refer to References.

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direct radiation and Parmelee's paper[2] on diffuse radiation. Yet there is a general lack of information available to the engineer on quantitative estimates of solar radiation during cloudless days.

During days with cloudiness even less information is available. Besides needing information on rates of direct and diffuse radiation during varying degrees of cloudiness, the engineer must know the distribution or frequency of occurrence of clear and cloudy days and the possible sequences of these days to be expected.

Through an experimental program of direct and diffuse radiation measurements, and through detailed examination of U. S. Weather Bureau radiation records, it is hoped that contributions to the knowledge of solar radiation availability will be made.

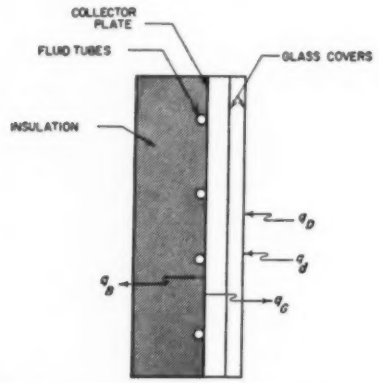

FIG. 1—SCHEMATIC FLAT-PLATE SOLAR ENERGY COLLECTOR

If reliable data on rates of direct and diffuse radiation are known, the performance of a flat-plate solar energy collector may be studied. Fig. 1 schematically shows the operation of such a collector with q_D the incident direct radiation, q_d the incident diffuse radiation, q_G the rate of heat loss from the plate through the glass covers, and q_B the rate of heat loss from plate through the insulation on the back side. For an ideal collector in steady-state operation with a uniform plate temperature t_p, the rate of heat collected per square foot of collector area is

$$q_a = (\tau\alpha)_D\, q_D + (\tau\alpha)_d\, q_d - (q_G + q_B) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

τ = the transmissivity of the glass covers.

α = the absorptivity of the blackened collector plate.

If

$$(\tau\alpha)_s = [(\tau\alpha)_D\, q_D/(q_D + q_d)] + [(\tau\alpha)_d\, q_d/(q_D + q_d)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

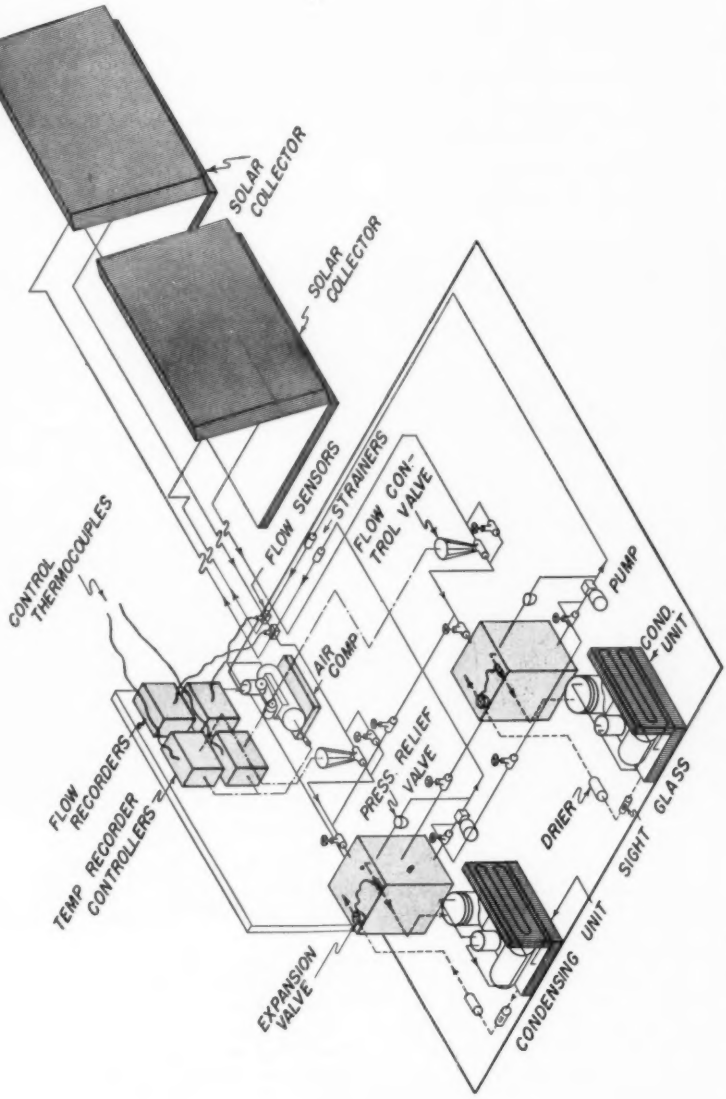

FIG. 2—MAJOR COMPONENTS OF EXPERIMENTAL COLLECTOR SYSTEM EXCLUDING TEMPERATURE MEASURING EQUIPMENT

then

$$q_c = (\tau\alpha)_t (q_D + q_d) - (U_G + U_B)(t_p - t_0) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

t_0 = outside temperature.

U_G and U_B = coefficient of heat transfer through the glass covers and the insulation on the back side, respectively.

The efficiency of collection is

$$\eta_c = q_c/(q_D + q_d) \quad . \quad (4)$$

FIG. 3—FRONT SIDE VIEW OF SOLAR COLLECTORS

Hottel[3] has presented data giving $(\tau\alpha)_t$ as a function of angle of incidence for a system of 2 or 3 glass covers with a blackened collector plate. An equation for calculating the loss factor U_G through the glass covers has been given by Hottel and Woertz[4].

Useful conclusions may be drawn from analyses of ideal collectors. Such analyses have been presented in a previous paper[5]. However, theoretical analyses are not sufficient to allow design of a flat-plate collector. Additional factors which must be studied include optimum tube size and spacing, optimum tube arrangement on the plate, and optimum fluid flow rate. The method of attachment of the tubes to the plate is important. The length of time required to warm a cold collector to operating temperature must be determined.

Adequate design information on flat-plate collector performance has not yet been published. In papers by Hottel[3] and by Hottel and Woertz[4], substantial contri-

butions have been made. G. O. G. Lof[6] has reported test information on collectors of the overlapped glass plate type.

At Minnesota a research program on flat-plate collectors is now in progress. This program includes a fundamental study of heat transfer from collector plate to fluid and a comprehensive experimental program on several flat-plate type collectors.

DESCRIPTION OF EQUIPMENT

Fig. 2 schematically shows the major components of the experimental collector system excluding the temperature measurement equipment. In general the sys-

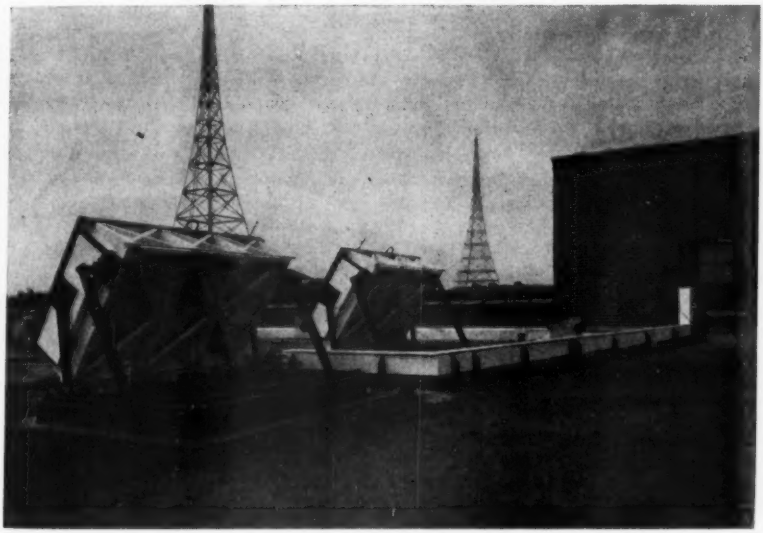

FIG. 4—BACK SIDE VIEW OF SOLAR COLLECTORS

tem consists of 2 collector units, 2 fluid transport circuits, 2 fluid-flow control systems, 2 insulated open fluid storage tanks, 2 circulating pumps, 2 refrigeration systems, and an air compressor for the pneumatic control systems.

Each collector unit consists of a wood exterior shell stiffened by 2 x 4 in. wood framing on the top, bottom, and back sides. Inside the shell is a 4 in. course of slab cork insulation. An insulated recess 7 ft-1 in. wide by 3 ft-3 in. high by 8 in. deep provides the space in which the collector plates are placed. A geared tilting mechanism allows positioning of the face of the collector in any position from horizontal to vertical. The collectors are mounted in a due south-facing position on the roof adjacent to the laboratory. Fig. 3 is a photograph showing the front of the collectors while Fig. 4 shows the back sides of the collectors.

Several collector plates have been designed for placing in the collector units. At present 2 plates with serpentine type coils placed on the front side of the plates have been installed. Two header-type coils will later be constructed. The 2 plates now in use are painted black and have ⅜ in. O.D. copper tubing spaced 6 in. on centers soldered to the 0.02 in. thick copper plate. The black paint used has an absorptivity for sunlight of approximately 0.96. Investigations are now in progress to determine the optimum type of paint and details in this regard will be reported later. Each collector plate has dimensions of 7 ft-1 in. wide by 3 ft-3 in. high. The copper plates are attached to a light wood frame which has a depth dimension of about 8 in. The construction of the 2 plates is identical except that the collector shown on the left in Fig. 3 has been designated as a control collector and its plate has more elaborate thermocouple provisions. Each collector plate is provided with immersion thermocouples located in the fluid lines immediately behind the plate for measurement of entering and leaving fluid temperatures. The plate used in the left unit of Fig. 3 has an additional 19 immersion thermocouples distributed throughout the length of the plate tubing and 25 thermocouples soldered to the back side of the plate. The plate used in the right unit of Fig. 3 has 4 plate thermocouples besides the 2 fluid-in-and-out thermocouples. The thermocouples of both plates are led to weatherproof connection boxes placed on the back sides of the units.

The glass covers are mounted in rubber and supported by a steel frame. The outer glass cover is tightly sealed in its rubber mounting while the rear glass cover is loosely mounted to provide breathing action for the space between the panes. A breather vent from the space behind the rear glass to the outside is provided. The spacing between the 2 glasses is approximately 1 in. and the rear glass is mounted about 1 in. in front of the collector plate. At present, ¼ in. *water white* (low iron content, high transmittance) plate glass is used as covers on both units. It is planned that these same glass covers will be used in the control collector for all experiments while different glass and probably plastic covers may be tested in the other unit.

The fluid piping, thermocouples, and wiring for the plate thermostats are led from the collector units back to the equipment room in a wood conduit filled with rock wool insulation. This conduit is shown in Fig. 4. Fluid connections from the copper tubing in the conduit to the collector units are made by use of flexible tubing to allow change of collector tilt. A system of by-pass valves at the end of the conduit next to the collectors is provided to allow recirculation of fluid in the outside lines back to the building before commencing operation in cold periods.

All equipment and instruments except the 2 collectors are located in a room immediately adjacent to the roof where the collectors are mounted. The location of this room relative to the collectors is shown in Fig. 4.

A 50-50 mixture by volume of ethylene glycol and water is used as the transport fluid. Major components of the fluid system located in the equipment room include 2 circulating pumps, 2 flow control valves, 2 flow measurement sensing elements and the 2 storage tanks.

Each circulating pump is a positive displacement impeller type rated at 1.76 gpm at 40 psi head pressure. As shown by Fig. 2, each pump may take its suction from either tank. A relief valve on the discharge side of each pump provides a by-pass back to the respective storage tank during times that the flow control valve is modulating the flow through the collector. Each fluid storage tank has a capa-

city of approximately 40 gal and is insulated with about 3 in. of glass wool. A refrigerant evaporating coil is placed in each tank.

Two refrigeration systems are provided for removing the energy absorbed by the transport fluid in the collectors. Each system uses dichlorodifluoromethane as the refrigerant and the condensing units are air cooled. One ¾ hp unit serves 1 tank and a ½ hp unit serves the other. The evaporators are helical coils made from ⅝ in. O.D. copper tubing and are immersed in the fluid in the storage tanks. Each evaporator is supplied refrigerant by a thermostatic expansion valve. As shown

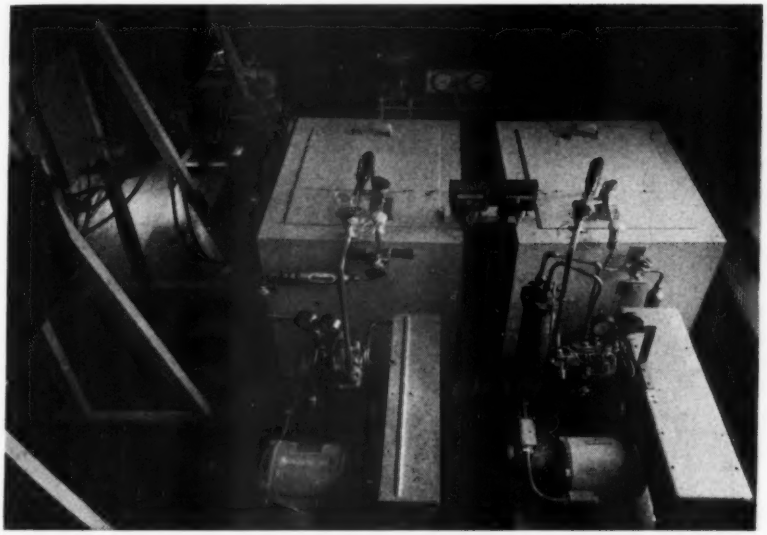

FIG. 5—VIEW OF REFRIGERATION SYSTEMS AND FLUID STORAGE TANKS WITH FLOW CONTROL VALVES AND AIR COMPRESSOR IN BACKGROUND

by Fig. 2, the transport fluid piping is arranged so one refrigeration system may cool the fluid from both collectors, or both refrigeration systems may be operated if one unit does not have adequate capacity. Fig. 5 shows a view of the refrigeration systems and the fluid storage tanks.

TEMPERATURE, FLOW AND RADIATION MEASUREMENTS

Instrumentation of the system was designed for recording of the most critical data so that a minimum of labor would be needed in conducting tests. Fig. 6 shows the recorder panel in the equipment room. The two units on the left are electronic-pneumatic controllers for operation of the fluid control valves. The 2 circular chart units next to the center of the panel are electronic recorders for measurement of fluid flow to the collector units. The 2 strip chart recorders on the

upper right end of the panel are for recording of the output of 2 Eppley pyrheliometers. The strip chart recorder at the lower right corner of the panel is a temperature recorder for 16 thermocouples. All recorders function on the electronic continuous balance principle.

In the control collector, 21 fluid thermocouples and 25 plate thermocouples are installed. In the other collector, 3 fluid and 4 plate thermocouples are installed. Including 2 thermocouples for outdoor temperature, a total of 55 temperature measurements at the collectors are possible. Of these, 16 may be automatically recorded and the remainder are read from an indicating potentiometer of the electronic self-balancing type.

Measurement of fluid flow is accomplished by employing a sensing element, a converter, and an electronic potentiometer. Fig. 2 schematically shows the loca-

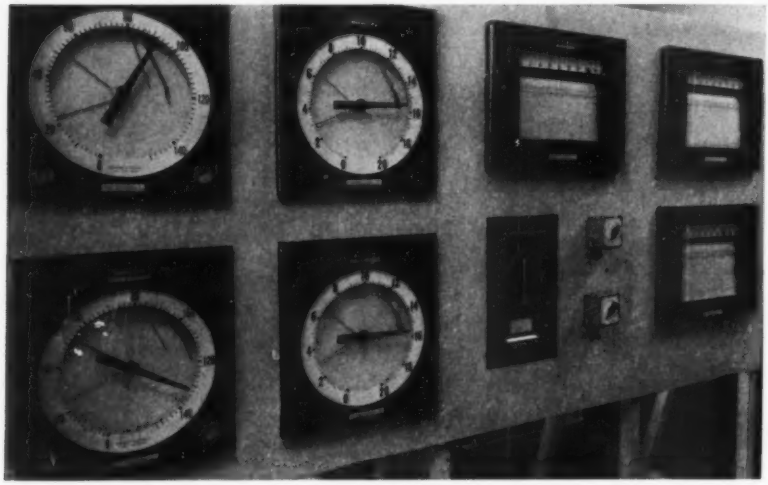

FIG. 6—RECORDER AND CONTROLLER PANEL IN EQUIPMENT ROOM

tion of the sensing elements. Each sensing element consists of a housing for direct mounting in the flow line containing a rotor which spins at a rate proportional to the fluid velocity. The sensing element produces an a-c output whose frequency is directly proportional to the rate of flow. The converter delivers a d-c voltage output proportional to the frequency of the a-c voltage of the sensing element. The potentiometer measures the d-c output of the converter and records it as flow.

During operation of the collectors, the incident radiation is measured by an Eppley pyrheliometer mounted at the side of the collector face parallel to the collector plate and in line with the center of the plate. Separate pyrheliometers are used for each unit to allow testing with different tilt positions. The d-c output of the pyrheliometers is recorded on strip chart recorders and the millivolt record is converted to Btu per (hour) (square foot) units. Calibration of the pyrheliometers was made at the ASHAE Laboratory at Cleveland.

In addition to use of the pyrheliometers in the collector studies, a separate research program on incidence of solar radiation is planned. Two pyrheliometer mounting stands with altitude and azimuth adjusting mechanisms have been constructed. Shading rings are now being built which will allow separate measurement of direct and diffuse radiation.

SYSTEM CONTROL

The collector system is designed for essentially complete automatic control during a single day's operation. This requires control of fluid flow and control of the operation of the refrigeration compressors, fluid pumps, and the air compressor.

In some experiments constant fluid flow through the collectors will be permitted but, in a majority of runs, modulating control of fluid flow to maintain a constant outlet fluid temperature at the collector or a constant plate temperature is planned. This control is accomplished by use of a pneumatically operated diaphragm motor valve placed in the fluid flow line and controlled by an electronic potentiometer linked to a pneumatic control unit. Fig. 2 schematically shows the location of the flow control valves while Fig. 5 shows a partial view of them.

Control of the refrigeration systems is accomplished through on-off operation of the compressors. A conventional type thermostat with a remote sensing element is wired directly in the compressor motor circuit. The sensing element is placed in the fluid storage tank next to the pump suction. The refrigeration system then operates to maintain an approximately constant temperature at the pump suction and thereby allows maintenance of an approximately constant fluid inlet temperature at the collector.

Operation of the air compressor for the pneumatic control system is controlled by a pressure switch which cycles the air compressor motor to maintain a constant receiver pressure.

Control of the operation of the fluid pumps is accomplished by use of plate thermostats. This control arrangement is a modification of a domestic electronic flow modulation system. Each system consists of a plate thermostat, an electronic relay amplifier, a switching relay, and a remote control point adjustment. The temperature sensitive resistance element of a wall-mounted thermostat was removed and attached to the back side of the collector plate. The wall thermostat case with its potentiometer and slider is mounted in the equipment room and the resistance element is wired back to it. The remote control point adjustment may be set at any temperature between 30 and 160 F. Each control system functions to start its fluid pump when the plate has reached the desired temperature and to stop the pump when the plate temperature falls below the desired value.

ACKNOWLEDGMENTS

A number of individuals and organizations have aided greatly in the design and development of these facilities. Both technical advice and help have been forthcoming from personnel of the University of Minnesota Mechanical Engineering Department not directly concerned with the project but who have assisted willingly on a number of occasions. Frank Traeger, Paul Dotray, William Owen, and John Peterson, students employed on the project, have contributed many valuable ideas and enthusiasm beyond that normally to be expected. In particular, acknowledgment is made to the generous assistance, both financial and technical, of Minneapolis-Honeywell Regulator Co., who made possible control and recording equip-

ment, in several cases unique, so necessary for the proper operation of the system and the project. E. F. Snyder, Jr., has been particularly helpful both through support and advice given on many different occasions. K. M. Lang, director of Engineering Research, Pittsburgh Plate Glass Research Laboratory, kindly arranged for the special glass covers used initially on the absorbers. G. V. Parmelee, C. M. Humphreys and E. R. Kaiser, of the ASHAE Research Laboratory in Cleveland, provided assistance both through valuable advice and through calibration of the pyrheliometers with special equipment which the Laboratory possesses.

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DISCUSSION

M. L. GHAI*, Evendale, Ohio (WRITTEN): I have read the paper with much interest and would like to make a few comments.

The laboratory is being designed apparently to emphasize study of engineering work on the utilization of solar energy. This is good. From my personal experience I have felt that there are 3 categories of institutions and individuals interested in the development of solar energy. First, there are institutions which are interested primarily in making outstanding discoveries such as finding inexpensive photo-electric materials capable of responding efficiently to the entire solar spectrum, or means of carrying out efficient photo-synthesis with or without the help of plants, or finding chemicals with large heating storage capacity. On the other extreme is the second category, mostly of individuals, where the enthusiasm is so great that there is no time or patience for elaborate experimental work. Thirdly, there are a few institutions which have followed systematic engineering approaches in an attempt to develop knowledge to handle solar energy in the best possible way without waiting for outstanding discoveries to be made. All 3 categories are important and need very much more encouragement than they have received so far. However, in the midst of eagerness to make outstanding discoveries, and enthusiasm to work fascinating equipment, the systematic engineering approach has been at a handicap. The laboratory being set up at Minneapolis should help this situation.

With careful selection of projects, the laboratory will, I hope, give engineers precise information to determine what kind of equipment is most suitable and economical for any given locality. The importance of this can be appreciated from the fact that, for

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* Rocket Engine Department, General Electric Company.

several applications, the use of solar energy is technically feasible even today; only the economic feasibility is doubtful. With precise engineering information, it would not be surprising to find that, for certain applications, the use of solar energy is even economically feasible for several sunny regions of the United States. For other areas, there will be better knowledge of the amount of progress necessary before the use of solar energy may be considered economical.

The authors have mentioned in the paper that the primary reason for establishing the laboratory is to get further information on the availability of solar energy and performance of flat-plate collectors. Our present knowledge about flat-plate collectors is fairly strong, although far from complete. There are other areas where the laboratory might find more fruitful results, for instance combinations of flat collectors and plane reflectors. I wonder if the authors plan the laboratory to be restricted to work on the conventional type of flat-plate collectors only.

R. C. LAST, Melbourne, Australia: I would like to ask Professor Jordan if, in his future studies on equipment, whether he will consider any variations in the mounting of the cell? If you could establish some mechanism to follow both the angle and tilt and orientation of the sun, do you think that such a study would produce an economical result, and that it may make the cell more efficient?

You appreciate the fact that the orientation of the universe in respect to the sun throughout the year, varies throughout the year.

R. W. MCKINLEY, Pittsburgh, Pa.: I share with the authors an interest in *improved* collectors, but I hope that they will include in their program the evaluation of presently operating solar collectors. I include homes, schools, and office buildings in this category and think that from a practical standpoint there is much more economic feasibility in these collectors than is generally realized. I hope that the authors may help us to evaluate these existing collectors, as well as improve upon them.

R. B. KNIGHT, Raleigh, North Carolina: I would like to ask Dr. Jordan if in his studies he anticipates going into the problem of storage of solar energy. This is a very important aspect of solar heating.

F. E. INCE, St. Louis, Mo.: I believe, Dr. Jordan, that you said constant temperature of the collector gives constant heat loss to the ambient. I believe this would assume a constant ambient temperature, wind velocity, radiant potential and other factors. I agree that the constant temperature level of the surface is a useful datum for comparative results. This may be difficult to maintain, but the outlet temperature level should be a feasible datum, and be most useful in comparative results.

AUTHORS' CLOSURE (Dr. Jordan): In answer to both Dr. Ghai's and Mr. Friend's comments, it is intended that our work involve studies of simple concentrators, albedo effect, and other similar factors beyond the study of simple collectors. We hope eventually to extend the work into comparisons of air and water collectors since the use of air collectors is indigenous to certain applications such as drying.

Insofar as our budget is concerned, some of our equipment has been purchased, some donated, and some loaned. Contributions to the work have been made through the research program of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS to the extent of probably a quarter of the total cost. We have felt both pleased and fortunate that the Society has contributed to our activities both financially and technically, and we hope that our activities can merit continuation of this support.

Concerning the comments of Mr. Last any practical applications of solar energy are, we feel, strongly involved with economic considerations. If we are concerned with heating and processing for the moment it would be, I am sure, economically unfeasible and architecturally difficult to attempt orientation of the collectors both seasonally and diurnally. Further, this is not as important with flat-plate collectors as it is with concentrating collectors where only direct radiation and no sky or diffuse radiation is

involved. Although several proposals have been made for rotating houses which would attempt to orient for maximum solar utilization, the thermal advantages are much more than offset by the mechanical disadvantages and initial costs, not to mention psychological considerations.

From the standpoint of collecting heat for processing, there are sizeable advantages involved in collector reorientation. If we are concerned only with the angle of tilt of the collector, the mechanical problems are not nearly as great as in collector rotation. The tilt angle in our collector studies is varied every few days, but for most applications probably changing the angle every one or two months is sufficient.

As Mr. Knight mentioned, energy storage is one of the important factors involved in the full utilization of solar energy. At the University of Minnesota we are not currently equipped for involvement in this work and prefer to concentrate our activities on those aspects of the subject more closely allied to mechanical engineering. It should be noted, however, that the ASHAE Technical Advisory Committee on Heat Pumps has already formed a special subcommittee to study all phases of thermal storage, since this is an important problem not only in solar energy utilization but in many other thermal problems.

I believe the point brought up by Mr. Ince was covered. There was no intent to infer that constant temperature of the collector plate would result in constant heat losses to the environment. This is certainly not true at low collector plate temperatures and only approximately true at very high collector temperature levels.

1584

No. 1584

INTERMITTENT GROUND GRIDS FOR HEAT PUMPS

By G. S. SMITH* SEATTLE, WASH.

SEVERAL studies have been made on the use of ground grids as a source of heat for the heat pump. Some were purely theoretical[1,2] studies using reasonable assumptions for the prevailing conditions and variables involved, others were partly or entirely experimental in nature,[3,4] and reported actual operating data. In comparison, these last included only a few tests in which the operation was intermittent to simulate operation under actual heating conditions.[5–8]

All of the studies have greatly assisted in an understanding of the various factors involved and have provided information useful for design purposes. However, because of the multitude of variables to be considered, many of which are difficult to determine, no method of design has been proposed that is not finally dependent upon a careful analysis of the proposed design in the light of all reported information from actual tests, or from successful installations. Perhaps no definite prescribed method of design will ever be found that will not depend to a large extent upon the use of such accumulated information. Thus such compilations of data would seem to be imperative, and should include as much detailed information of existing conditions, design factors, and even operating results as would seem to bear directly upon the problem.

Studies, both theoretical and experimental, have been made at the University of Washington, and some of the results have been reported.[4,9] A more recent series of tests were made during the heating season of 1952–53 in which the heat pump operation followed the intermittent heating requirements demanded by the weather.

INTERMITTENT TESTS AT UNIVERSITY OF WASHINGTON

To determine the time for operating the heat pump each day, the total operating time for each month was calculated and a portion of this time allocated to each day in such a way as to cover a wide variety of running periods, from a very brief

* Professor in Electrical Engineering, University of Washington.

† Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956.

run to almost continuous operation for the whole 24 hr of the day. The monthly operating time was based upon the predicted average coldest day of the year. This average coldest day was determined by taking the degree days for the average coldest day for a number of years picked at random from the last 50 years of records.

FIG. 1—SINGLE EXPOSURE OF PHOTOGRAPHED DATA

Assuming that the heat pump would operate almost continuously on such a day, the monthly operating hours were calculated as follows:

$$\text{Operating hours per month} = \text{Hours in month} \times \text{percent of operating time.}$$

The percent of operating time is the ratio of the average degree days for that month divided by the product of the days of the month and the degree days of the average coldest day as just mentioned. This will closely approximate the average intermittent operation under actual heating conditions.

The ground grid used for these tests consisted of one loop of ½ in. O.D. copper tubing, with an effective heat collecting length of 150 ft. It was placed 4 ft below the surface, with the 2 sides of the loop spaced 4 ft between centers.

The soil was sandy with some gravel and a little fine material approaching the nature of clay. Its casa grande classification would be S.W. with a dry weight of 97 lb per cu ft. Other details of the soil characteristics are given in No. 8 of Table 4.

A 2-hp motor and suitable compressor was used as a heat pump. Dichlorodifluoromethane was expanded directly into the ground grid by means of a thermal type of expansion valve. All important temperatures and pressures, as well as the refrigerant flow while in its liquid stage, were photographically recorded every 5 min. The average of the 12 readings per hr was used for computations. Fig. 1 shows an enlarged print of one of the frames of the records taken.

The scheduled program of extracting heat was started November 1 but continuous records were not started until the recording apparatus was finally made

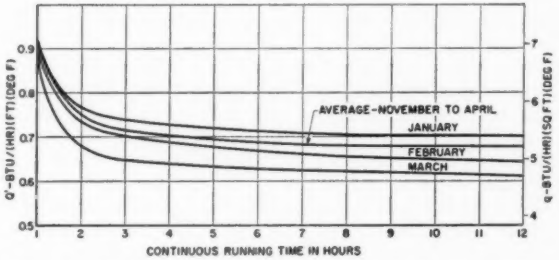

FIG. 2—HEAT TRANSFER CHARACTERISTICS OF GROUND GRID FROM UNIVERSITY OF WASHINGTON INTERMITTENT TESTS

available on December 1. The run was continued through until the last of March except for approximately 1 week in December when, because of a burned out bearing, a new compressor was installed. The effectiveness of the heat withdrawn was compensated for by running longer periods for the remainder of the month.

RESULTS FROM INTERMITTENT TESTS

The relation of the average final value of Btu/(hr) (ft) (F deg), which will be termed Q', and the length of the run t is shown by the curves in Fig. 2, as determined by all the various runs taken during each month. The temperature difference was taken as the difference between the average refrigerant temperature in the grid as an evaporator, and the undisturbed soil temperature at the same depth and about 25 ft away so as not to be affected by the grid. The curves for each succeeding month were lower than those of the previous month because the average temperature of the general soil volume was decreasing. This trend would reverse as soon as the heat due to the spring weather could penetrate to the grid depth. The minimum undisturbed soil temperature at a depth of from 4 to 6 ft usually occurs from the latter part of February to the first part of March.

The curve for the averages over the heating season included the higher values

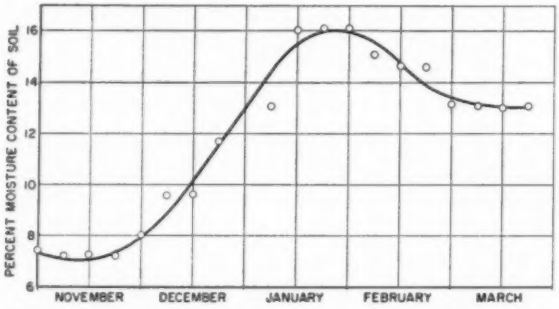

FIG. 3—SOIL MOISTURE TAKEN AT FOUR FOOT DEPTH UNIVERSITY OF WASHINGTON INTERMITTENT TESTS

for November and December which were not derived in enough detail to be included as separate curves.

Fig. 3 gives the average soil moisture at the grid depth during the season. The autumn weather was rather dry, but the rainfall during January was unusually high. With rainfall normal for November and December, the earlier curves would have been somewhat higher. Rainfall during the heating season adds greatly to the available ground heat. During a rain the air, and thus the water temperature, is always well above freezing, and because of its high specific heat the water carries a considerable amount of heat into the soil. Soil moisture was measured by commercial glass fiber soil-moisture cells.

As the run progressed, a cylinder of frozen soil formed about the tube and increased to a radius of about 6 in. There is a tendency for frozen soil to slightly increase its thermal conductivity as the temperature decreases. This frozen cylinder would then tend to increase rather than decrease the heat withdrawal effect.

In addition to this the frozen cylinder provides an apparent heat storage effect, since at freezing temperature the latent heat of freezing is released with no change

TABLE 1—GENERAL DATA ON GRIDS USED IN INTERMITTENT RUN TESTS IN TABLES 2 TO 5.

TESTS BY	TUBE O.D., IN.	OUTSIDE TUBE AREA, SQ FT/FT	ACTIVE TUBES		SPACING FT	REFRIGERANT USED[a]	SOIL NO.	DEPTH BELOW SURFACE FT
			LENGTH IN FT	NO. IN PARALLEL				
Univ. of Washington	½	0.13	150.1	2	4	A	8	4
Univ. of Kentucky	1⅛	0.294	170	1		B	7	5 to 5.6
Univ. of Kentucky	1⅛	0.294	317.1	9	4.75		7	4 to 5
Comm. Edison Co.	1⅝	0.425	155	2	10	A	9	6
Phil. Elect. Co.	1²⁹/₃₂	0.500	Variable	2	1.25	B		5

[a] A = dichlorodifluoromethane; B = antifreeze mixture.

TABLE 2—AVERAGE HIGH AND LOW VALUES OF SEVERAL OF THE IMPORTANT FACTORS REPORTED ON THE VARIOUS TESTS SPECIFIED IN TABLE 1.

	TESTS BY	YEAR	HOURS RUNNING TIME	NORMAL SOIL TEMP, F	AVG REFRIG. TEMP, F	Q' BTU PER (HR) (FT) (F DEG)	q BTU PER (HR) (SQ FT) (F DEG)	OTHER FACTORS
Avg	Univ. of	'52	10.5	49.5	6.66	0.707	5.43	
High	Wash.	and	24	51.0	8.58	0.799	6.15	
Low		'53	4	49.0	3.82	0.595	4.58	
Avg	Univ. of	'49	12	46.0[a]	23.1	2.14	7.28	Line
High	Kentucky	and	12	60.9	35.2	5.98	20.3	Type
Low		'50	12	40.6	16.5	1.14	3.88	Source
Avg	Univ. of	'49	12	46.0[a]	23.9	0.784	2.67	Grid
High	Kentucky	and	12	60.9	33.8	1.64	5.58	Type
Low		'50	12	40.6	17.2	0.425	1.45	Source
Avg	Univ. of	'52	12	47.4[a]	13.7	0.936	3.19	Line
High	Kentucky	and	12	62.0	27.1	1.28	4.35	Type
Low		'53	12	38.4	8.0	0.614	2.05	Source
Avg	Univ. of	'52	12	47.4[a]	14.4	0.420	1.43	Grid
High	Kentucky	and	12	62.0	24.5	0.647	2.20	Type
Low		'53	12	38.4	0.88	0.318	1.08	Source
Avg	Comm.	'47	10.3	51.0	19.5	2.08	4.9	
High	Edison	and	17.5	60.2	23.0	3.15	7.65	
Low	Co.	'48	1.0	45.4	17.6	1.40	3.30	
Avg	Phila.	'48		52.4	41.0[b]	1.60	3.20	Actual
High	Elect.	and		64.7	70.3	6.60	13.20	Residence
Low	White-marsh	'49		45.4	29.6	0.44	.88	Heating
Avg	Phila.	'48		52.9	40.0[b]	1.46	2.92	Actual
High	Elect.	and		64.5	64.0	3.10	6.20	Residence
Low	Lands-downe	'49		47.5	28.0	0.83	1.66	Heating

[a] Normal temperatures given here were at 18 in. below coil.
[b] Average refrigerant temperatures were taken at the surface of the ground coil in this case.

in temperature gradient thus adding to the heat withdrawn while the pump is running. During the time when the heat pump is not in operation this somewhat enlarged cylinder continues to extract heat from the surrounding warmer soil, replacing the latent heat with no change in temperature until the latent heat is all restored and the soil is no longer frozen.

The curves in Fig. 2 show a rapid decrease in Q' during the first 2 or 3 hours of operation but after that time the heat flow into the tube approaches a more nearly constant value. The heat extraction factor Q' is below that calculated by theory, as can be seen by comparing the Q' at 10 hr in Fig. 2 with that for 10 hr in Fig. 5, on curve for University of Washington Soil Sample No. 8.

This 150 ft loop of grid supplied a minimum of about 4000 Btu per hr. When the soil was reasonably moist due to the rainfall in January, the rate ranged from 4300 to 5000 Btu per hr. With a spacing greater than 4 ft the grid would have supplied a small amount more, but at a smaller spacing the decrease would have

been even more noticeable. With several tubes in parallel at the same depth and spacing, a further decrease would result. However, the tests indicate that the earth does provide a dependable source of heat providing enough volume is drawn upon not to decrease the average temperature of the volume below 35 or 40 F. Freezing any large percentage of the soil volume will not greatly increase the amount of heat obtained and may prove very destructive if building foundations are nearby.

COMPARISON WITH SIMILAR REPORTED TESTS

Since considerable detailed information on 3 other similar tests was available, studies for comparing results were made. The complete set, or even an abbreviated set, of the computed values for comparison cannot be given here. However, Table 2 does give the overall average, high and low values for each set which at least indicates a fair correlation of results considering the diversity of the factors involved such as the tube size, soil properties, and refrigerant temperatures.

TABLE 3—DATA FROM MISCELLANEOUS TESTS BOTH CONTINUOUS AND INTERMITTENT OVER LIMITED PERIODS

TESTS BY	TUBE O.D. IN. DIAM.	LENGTH FT	DEPTH FT	SPACING FT[a]	SOIL TYPE	AVG TUBE TEMP	AVG SOIL TEMP	HOURS RUN	Q' BTU PER (HR) (FT) (F DEG)	q BTU PER (HR) (SQ FT) (F DEG)
Univ. of Conn.	⅞	150	6			30.0	43.8	645	2.29	10.0
	⅞	150	6			15.0	47.0	719	1.70	7.4
	1⅜	150	8			15.5	41.7	600	2.10	5.8
Kan. State Col.	1⅛	100	6	2	Sandy	35.0		400	1.68	5.7
	1⅛	100	6	10	Sandy	21.0		400	2.42	8.2
	1⅛	100	6		Sandy	20.8		670	1.16	3.9
	1⅛	100	6		Clay	30.0		6-on 6-off	1.25	4.2
	1⅛	100	6	A	Sandy	10.0		6-on 6-off	1.42	4.8
	1⅛	100	6	A	Clay	10.0		6-on 6-off	1.45	4.9
	1⅛	100	6	A	Sandy	10.0		163	1.01	3.4
	1⅜	100	6	A	Clay	10.0		Cont.	1.31	4.4
Southern Research Institute	1¼	88	3	A	Sandy	34		200	2.55	8.6
	1⅜	88	3	A	Clay	34		200	2.15	7.3
Texas A. and M.	½	160	5	10				Variable	1.01	7.7
Util. Res. Commission	1¼	155	6	10	Clay	18	43	Variable	1.70	4.0
	1⅜	155	2	10	Clay	18	43	Variable	1.40	3.3

[a] A = ½-in. tube inside of 1¼-in. tube.

In obtaining the values for University of Kentucky, the so-called normal soil temperatures were taken as the temperature 18 in. below the coils since this appeared to be the best temperature at any distance that was given. The actual normal soil temperature was doubtless somewhat higher in general. In these tests the grid source gives a considerably lower rate of heat transfer than for the line source. The greater values of heat transfer for 1949–50 as compared to those for 1952–53 might be accounted for by the greater rainfall in the 1949–50 season.

For the Philadelphia Electric results, the evaporator coil temperature was used instead of the actual refrigerant temperature because the latter was not given.

In general the high and low values are often extreme because they may have been unusual, such as records at the start before average conditions prevail, or

under abnormal conditions existing at times but not observed or at least reported. The average values should, however, be representative for such comparisons.

Table 3 gives additional data of a similar nature. In most cases these tests were of single continuous runs of various time durations, but they show very similar results. Tables 2 and 3 give only a small portion of all the results available, but a more extensive set of data would reveal very little additional information.

Determination of all the reasons for the variations in the tests considered would probably be next to impossible. However, further study of a few of the major

TABLE 4—SOIL PROPERTIES FROM VARIOUS REPORTS

No.	Location	Casa Grande Symbol	Soil Type	Liquid Limit	Plastic Limit	Dry Density pd	Spec. Heat, S	Thermal Diffusivity, α	Moist. % Dry wt	Therm. Cond. K	Reference
1	Columbus, Miss.	S.C.	Clay Sand			107.1	0.163	0.029	5	0.66	10
								0.038	10	1.06	
								0.045	20	1.74	
								0.046	30	2.28	
2	Cleveland, Tenn.	O.L.	Sandy Silt	28.6	23.4	94.3	0.149	0.018	5	0.33	10
								0.022	10	0.52	
								0.024	20	0.80	
								0.022	30	0.94	
3	Bristol, Va.	M.H.	Silty Clay	55.1	39.6	75.3	0.193	0.011	5	0.17	10
								0.012	10	0.27	
								0.015	20	0.43	
								0.016	30	0.60	
4	Jackson, Tenn.	C.L.	Lean Clay	46.7	26.3	84.5	0.202	0.012	5	0.26	10
								0.014	10	0.35	
								0.017	20	0.59	
								0.018	30	0.76	
5	Decatur, Ala.	C.L.	Lean Clay	29.0	21.5	86.3	0.176	0.018	5	0.38	10
								0.024	10	0.57	
								0.024	20	0.85	
								0.024	30	0.91	
6	Paris, Tenn.	C.L.	Lean Clay	49.6	25.2	83.3	0.188	0.011	5	0.22	10
								0.013	10	0.31	
								0.017	20	0.55	
								0.018	30	0.74	
7	Lexington, Ky.	C.L.	Lean Clay	43.4	27.3	95.0	0.20	0.019	26	0.84	10
8	Seattle, Wash.	S.W.	Sand Gravel			97		0.026	14	0.73	11
9	Comm. Ed. Ill.					110		0.0257	18	1.07	5
11	Dakota Co., Minn.	S.M.	Sandy Loam			110			4	0.45	5
									10	1.08	
12	Ramsy Co., Minn.	C.L.	Sandy Loam			110			4	0.54	5
									10	0.83	
13	Northway, Alaska	M.L.	Silt Loam			90	0.176		10	0.33	5
									20	0.50	

factors involved might be of value. Results of these additional studies will be presented with the hope that they will be of value to designers and perhaps inspire more extensive compilations of results, as well as new and more extensive studies.

SOIL PROPERTIES

If the designer could arrive at some value of thermal conductivity, K, of the soil about the grid with no complications due to surface films or other irregularities, and the soil body could be assumed an infinite uniform medium, his task would be comparatively easy. Even then, to arrive at a reasonably accurate value of K would probably require tests or at least a good knowledge of K values for similar soils.

A considerable amount of careful and detailed work has been done on the thermal properties of various soils under many varying conditions. A rather extensive and detailed compilation of a major portion of such reported work would enable a designer to select a K value for his soil as accurate as is necessary for his purposes. He must still determine the soil type, the approximate natural density, and the average moisture content. A very brief table of such results is presented in Table 4, in which references to the sources are given. By referring to these and other reports, a more extensive set could be easily assembled.

From this table it is apparent that the soil conductivity varies from about 0.25 to around 2.25, with the greater portion running around 0.75 to 1.00. The effect of moisture is very evident. Sand is usually superior to clay since solid quartz has a higher thermal conductivity than water. Dry sand is poor because of the air

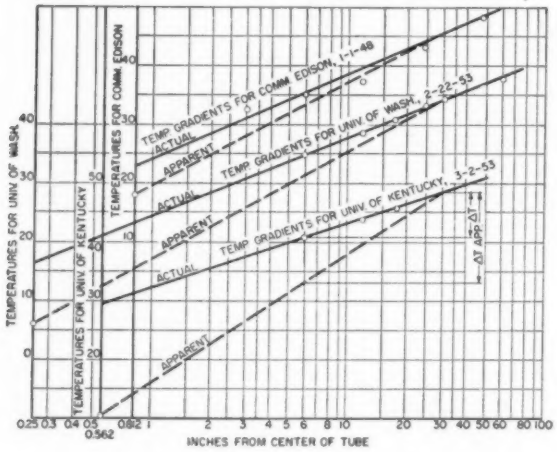
TEMPERATURES FOR UNIV OF WASH.
TEMPERATURES FOR UNIV OF KENTUCKY
TEMPERATURES FOR COMM. EDISON
TEMP GRADIENTS FOR COMM. EDISON, 1-1-48
ACTUAL
APPARENT
TEMP GRADIENTS FOR UNIV OF WASH., 2-22-53
ACTUAL
APPARENT
TEMP GRADIENTS FOR UNIV OF KENTUCKY, 3-2-53
ACTUAL
APPARENT
Δt APP. Δt
0.25 0.3 0.4 0.5 0.562 0.82 1 2 3 4 5 6 8 10 20 30 40 50 60 80 100
INCHES FROM CENTER OF TUBE

FIG. 4—TEMPERATURE-GRADIENT PLOTS FOR STUDIES ON FILM COEFFICIENT EFFECTS

voids between grains, but dry clay is even poorer than sand. In general clay will hold large percentages of water, tending to give it a K factor dependent largely upon the moisture present. Sand will usually hold sufficient moisture to effectively fill the voids. The moisture of soils at usual grid depths is normally almost constant, but varies noticeably with long dry or unusually wet spells.

FILM COEFFICIENTS

In the design of a heat exchanger the effect of the infinitesimally thin films just inside and just outside the metal surface must usually be taken into account. Such effects are also present in ground grids but have usually been given little or no attention. Film coefficients are in general very difficult to determine and often vary over wide ranges. In this respect the ground grid is no exception. During

earlier work at the University of Washington some special tests of this nature were made and mentioned in reference 4. The results of these tests were far from satisfactory and were not included. However they definitely indicated film effects present and compare reasonably well with the studies to be given here.

The theoretical temperature difference between any 2 points outside of a heated or cooled tube in an infinite uniform medium and where no end effects are present can be expressed by the equation.

$$T_2 - T_1 = [U \log (r_2/r_1)/2\pi lK] = [Q' \log (r_2/r_1)/2\pi K] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

Thus a plot of actual temperature at varying distances from the center of an isolated tube should be a straight line if the distances are plotted on a log scale against temperature on a uniform scale. Solving Equation (1) for K indicates that the thermal conductivity of the soil can be determined and that it is proportional to the slope of the temperature difference between 2 selected points on the straight line curve.

Most of the intermittent tests studied contained sufficient measured temperatures and other information to enable the plotting of such curves. Three of such curves are shown in Fig. 4. The direction of the measured temperatures was taken to avoid the effect of adjacent grids as much as possible. For the University of Kentucky, the temperatures at 6, 12, and 18 in. directly below the grid were used. In others, the temperatures at horizontal distances at the grid level in a direction away from other tubes were used.

By projecting the straight line to the tube radius in the plot, the temperature difference between this intersection and the average refrigerant temperature within the grid is a reasonably accurate indication of the film effect. With q known the film coefficient h (inside plus outside) will be

$$h = q/t_f = \text{Btu per (hr) (sq ft) (F deg)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

To use this factor the designer must calculate the heat flow due to a composite temperature difference made up of the temperature across the tube films using the film coefficient h involving zero distance, and the resulting temperature gradient within the soil with its thermal conductivity factor K. Perhaps an easier method could be proposed that would be almost as accurate in most cases by using a value of K decreased in value to account for the film effect. Such possible multiplying factors will be derived as follows.

By selecting 2 distances from the center of the tube, say 6 in. and 30 in., the temperature difference on the solid line, which is the actual temperature gradient within the soil, can be used in Equation 1 to find the true K for the soil as indicated by this actual temperature gradient. By drawing a straight line from the 30-in. point to the average refrigerant temperature at the tube radius, the temperature difference between 6 and 30 in. can again be found on this straight line. This is shown by the dash lines in Fig. 4. Using this apparent temperature difference in Equation 1, an apparent value of K can be found which will be designated K_a. The ratio of K_a to K will give the ratio by which the K of the soil should be decreased to give results equivalent to the actual conditions. The distance of 30 in. was chosen as a reasonable distance beyond which the soil temperature would probably be constant, regardless of the grid operation. Because of the nature of the log scale used for distances, doubling the 30 in. dimension would produce only a very small decrease in the apparent slope thus derived. The choice of 30 in. rather than 60 in. or more results in a greater factor of safety in design constants.

Table 5 gives a number of both K_a/K and h values determined by this method. Each set was calculated from the data reported for a given day of operation, and again selections were made at random, or in some cases, only where sufficient data

TABLE 5—RESULTS FROM STUDIES OF EFFECT OF TUBE SURFACE FILMS

DATE	THERMAL CONDUCTIVITY		$\frac{K_a}{K}$	FILM Δt	FILM COEF. h IN BTU PER (SQ FT) (HR) (F DEG)	AVG TUBE TEMP
	ACTUAL K	APPARENT K_a				
FROM UNIVERSITY OF KENTUCKY TESTS						
11–4–52	1.030	0.485	0.470	27.0	2.8	18
12–2–52	0.790	0.430	0.545	24.3	3.5	14
1–15–53	0.603	0.345	0.572	19.2	3.16	13.5
2–13–53	0.618	0.328	0.532	21.6	2.84	9.3
3–2–53	0.637	0.344	0.540	19.7	3.16	10.4
4–1–53	0.900	0.420	0.466	20.3	3.38	14.0
FROM COMMONWEALTH EDISON TESTS						
12–10–47	2.17	1.68	0.773	8.5	21.8	19
1–1–48	1.62	1.41	0.870	5.0	30.0	18
2–1–48	1.25	1.19	0.950	1.5	72.5	19.4
2–19–48	1.78	1.47	0.827	1.0	122.0	20
3–1–48	1.69	1.62	0.960	1.5	94.0	18.9
4–1–48	1.62	1.50	0.923	2.3	52.8	20.0
4–10–48	2.14	1.91	0.937	2.6	55.5	20.0
FROM UNIVERSITY OF WASHINGTON TESTS						
1–16–53	0.915	0.715	0.780	9.95	24.9	8.0
2–1–53	0.785	0.660	0.840	9.56	25.1	7.3
2–22–53	0.825	0.633	0.767	10.40	22.0	6.0
3–5–53	0.770	0.602	0.780	9.20	23.6	5.50
3–19–53	0.703	0.585	0.833	8.0	27.5	5.23
3–27–53	0.670	0.604	0.900	6.85	33.7	5.50
FROM PHILADELPHIA ELECTRIC TESTS WHITEMARSH						
2–22–48	1.100	0.94	0.855	1.7	18.3	35.9
11–30–49	0.010	0.94	0.945	1.5	37.0	30.9
12–21–49	0.995					
2–21–50	1.240					
LANDSDOWNE						
12–20–49	1.33	1.11	0.835	1.6	54.0	29.7
11–15–49	1.24	1.00	0.810	1.7	20.0	50.0
1–17–50	1.08					32.0
2–21–50	1.42	1.09	0.84	2.0	29.1	26.9

were available. As might be expected from comparisons of previous results, these results are variable. While no claim can be made that the method or results are highly accurate, the study does definitely indicate the need for some means of

accounting for this factor, and the figures presented can serve as a guide in estimating the effect. More careful studies on this feature would be of considerable value in the design of ground grids.

EFFECT OF TUBE SIZE

The effect of the tube size on the amount of heat extracted has often been mentioned but to the author's knowledge few if any specific figures or curves have been presented. For that reason a set of several types of soils were selected, including those for 3 of the tests reported, and the values of Q' and q were calculated. Cal-

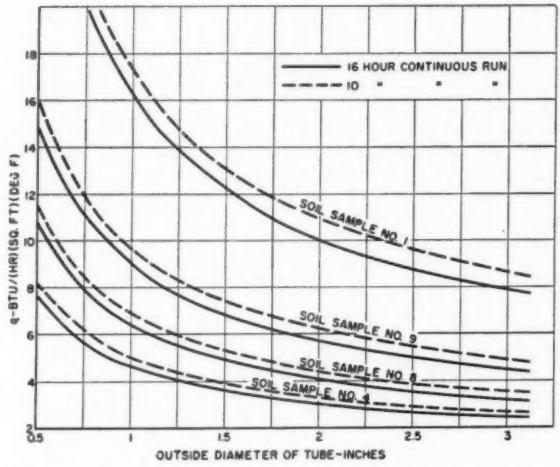

FIG. 5—CURVES SHOWING RELATION OF OUTSIDE DIAMETER OF TUBE TO ITS HEAT TRANSFER FACTOR q CURVES CALCULATED BY MEANS OF THEORETICAL EQUATIONS

culations were made using Ingersoll's method, assuming a single isolated tube. Sizes from ½ in. to 3 in. were included, and continuous runs of 10 as well as of 16 hr were used. Fig. 5 gives the results in curve form for q plotted against the tube sizes. The Q' value is merely the product of q and the external square foot area of the tube per foot of length.

These plotted values are higher than those obtained by tests in Tables 2 and 3 because ideal conditions were assumed. However the relative values for the various tube sizes should be accurately indicated by the curves.

Evidently the smaller tubes are more effective per unit of area than the larger ones. This is not surprising since the soil volume drawn from is not appreciably different, and the larger area presented by the larger tube becomes less effective per unit of area. However, the pressure drop of the circulating liquid or gas must be maintained at a reasonable value, and this can be accomplished either by the use of a larger tube or by several tubes used in parallel. The latter choice often

gives trouble in distribution of flow and is usually avoided or overcome only by the use of an individual expansion valve or a circulating pump for each of the parallel circuits.

EFFECTS OF SPACING OF TUBES IN GRID

Still another factor that has been given too little attention is the effect of proximity of the tubes in the grid. A set of calculated curves is presented in Fig. 6. The plot shows 4 sets of curves where the full-line curve is calculated for 3 parallel tubes in a grid, while the broken-line curve is for an infinite grid. Most actual grids will fall between these two. Each set was calculated for a specified ratio of spacing between the centers of tubes to the radius of the tube. The coordinates are the ratio of Q'_n/Q'_1 *vs* $A\ t$. Here Q'_n is for a grid consisting of n tubes in parallel,

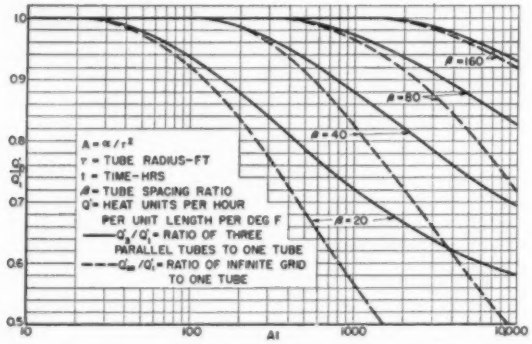

FIG. 6—HEAT TRANSFER COEFFICIENT RATIOS CURVES CALCULATED BY MEANS OF THEORETICAL EQUATIONS

while Q'_1 is for a single isolated tube. The constant A is equal to α/r^2 where α is the diffusivity of the soil and r the radius of the tube in ft, while t is time in hours of continuous operation. Such curves can be used with any type of soil or for any tube size. By having these 2 factors specified, the time in hours at any point on a specified curve is easily determined with reasonable accuracy. However the relation between line and grid sources given in Table 2 appears to give greater differences than were indicated by the calculated curves, and here again more information from actual tests would be desirable.

USE OF INFORMATION FOR PRELIMINARY ESTIMATES OR DESIGNS

No design procedure has been suggested since this could well be the topic of a more extensive paper, and because several proposals have already been made.[4, 13, 14] However, a suggestion as to the use of such compiled information, as is briefly presented here, in making preliminary estimates on a grid design might be of value. With a well selected compilation of such data to cover almost any proposed set of conditions, such methods may later prove to be adequate for final design.

Problem: Assume a residence requiring 50,000 Btu per hr, using a heat pump with a C.O.P. of 3. The necessary heat to be extracted from the heat source would be 2/3 of the

total heating load or approximately 34,000 Btu per hr. Let further assumptions be as follows: soil type that of No. 2 in Table 4, having an average moisture content of 21 percent, and thermal conductivity, K, of 0.73 at the final grid depth; evaporator grid made of 1 in. outside diameter tubing; probable lowest mean soil temperature in February of 40 F; average refrigerating temperature, to promote some freezing, 12 F; average temperature difference, 28 F.

Solution: The soil type and other conditions correspond fairly closely to the University of Washington tests with an average q of 5.43. Since this was obtained with a ½ in. diameter tube, the ratio of q for a ½ in. to a 1-in. tube after running 10 hr can be found in Fig. 5 and is approximately 0.6. Thus the 1-in. tube would give a q of about 5.43 × 0.6 or 3.26, and since its outside area is 0.262 sq ft per ft the Q' value will be 0.854. Since the temperature difference was 28 deg, the Btu per hr per ft of tube is 23.9. Thus the 34,000 Btu heating load would require about 1430 lineal ft of tubing if so spaced that they do not appreciably affect each other.

Assuming a 4-ft spacing the mutual effect between adjacent tubes could be approximated by using Fig. 6. Thus β is 48/0.5 = 96; while A ($= \alpha/r^2$) will be $0.24/(0.5/12)^2 =$ 14 approx. The continuous running time before this spacing should affect the operation would be about 1000/14 = 75 hr. This raises the question as to whether 75 hr would simulate the effective previous intermittent operation, and is one factor which still needs further study. At least about 140 hr would be needed to appreciably reduce the effectiveness of the tube. With this spacing the mutual effects should be rather negligible.

With a 4-ft spacing, assuming 100-ft runs, the number of parallel tubes required would be approximately 14 or a plot 56 ft wide, and thus 5600 sq ft of soil are required. A more careful design should be used to determine whether it need be somewhat larger or could be somewhat smaller, as well as determination of the desirable spacing and depths.

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CONCLUSIONS

There seems little question but that the intermittent type of tests gives more dependable information for use in practical design of ground grids than that obtained from single continuous runs. A greater number of tests, each with a considerable amount of details of design factors, measured temperatures, etc., would be most desirable. More tests for comparing the operation of grid and line sources as well as tests to determine the nature and effects of film coefficients, would provide valuable and much needed data.

NOMENCLATURE

U = Btu per hour.
Q = Btu per (hour) (foot)
Q' = Btu per (hour) (foot) (Fahrenheit degree)*.
q = Btu per (hour) (square foot) (Fahrenheit degree)*.
T = temperature Fahrenheit.
r = tube radius in feet.
r_1 = distance in feet from center of tube to point of temperature T_1.
r_2 = distance in feet from center of tube to point of temperature T_2.
l = tube length in feet.
K = thermal conductivity—Btu per (hour) (foot) (square foot) (Fahrenheit degree).
α = thermal diffusivity = K/c_p—square feet per hour.
ρ = density—pounds per cubic foot.
β = spacing ratio = Z/r.
Z = spacing between tube centers in feet.
t = time in hours.
h = film coefficient—Btu per (square foot) (Fahrenheit degree) (hour).
n = number of parallel tubes used in a grid.

DISCUSSION

B. J. FLUKER†, College Station, Texas (WRITTEN): Professor Smith is to be commended for his investigation concerning the intermittent operation of the heat pump ground coil. The operation of an actual heat pump installation would require, as the author has pointed out, intermittent operation rather than the idealized continuous running to maintain a desired temperature level of the conditioned structure.

The investigation reported here attacks the problem on the basis of operation following the intermittent heating requirements per month demanded by the weather. The computation procedure employed to determine the average number of heat pump operating hours per month appears to be sufficient to closely approximate a practical installation. After these computations were made, it is assumed that the author al-

* Fahrenheit degree temperature difference is that between refrigerant in evaporator and the normal undisturbed soil except for the Philadelphia Electric Co. tests where the tube temperature only was given.
† Texas Engineering Experiment Station.

located to each successive day of the month a number of hours of heat pump operation in relation to whether the ambient air temperature was decreasing, as at the beginning of the heating season, or increasing, which is the case near the end of the heating season. That is, the number of hours operation on successive days should have increased as the average daily ambient air temperature decreased toward its annual minimum while the reverse is true when the ambient air was approaching its maximum temperature. This would also closely simulate actual operation.

Analyzing still further the periods of heat pump operation employed, it is indicated that the total hours allocated to any single day were run continuously during that day. It is suspected, however, that this would very seldom, if ever, be the case in a comfort conditioning heat pump application. Repeated cycling operation, such as a percentage of unit time *on* followed by the remaining portion of unit time *off*, would most likely be evident in actual installations. The unit of time would perhaps be no greater than 3 to 4 hours except possibly on very mild days which would require almost no heating. In all probability the cycle would be of even shorter duration over the majority of the heating season.

Three factors which would regulate the percentage of unit time the system would operate are

(a) varying ambient conditions which determine the actual heat loss at any time of the structure being conditioned,

(b) the ratio of the actual heat loss of the structure at any time to the heating capacity of the system, and

(c) the tolerance within which the desired temperature level of the structure must be maintained.

The foregoing discussion leads to another point. Were the intervals between periods of heat pump operation in this study sufficiently long to permit recovery of soil temperatures to their natural undisturbed values? If this be the case could not each running period be considered as an individual continuous operation run of short duration? The description of the tests indicated this to be possible after the shorter runs. Conversely, if the time between running periods, such as has been observed in actual operation, allowed only partial recovery of the surrounding soil temperatures, the rate of heat gain at the beginning of the first operation period would be greater than the rate for the second period, the second greater than the third, and so on, until a constant rate is reached or the cycle of operation changes. The decrease in each successive period would not be as great as that corresponding to continuous operation at the same accumulated time. The overall effects of incomplete soil temperature recovery accompanying relatively short intermittent cycling would be

(a) to smooth the inverse knee which appears in the curves of Fig. 2 of the paper,

(b) to increase the time required to reach a constant rate of heat transfer between soil and coil,

(c) to increase the constant rate of heat transfer ultimately reached, and

(d) to increase the ultimate rate of heat transfer in proportion to the percentage of unit time the heat pump is operating.

The discussion further emphasizes the conclusion as brought out by the author that the continuous type test alone is insufficient to provide a basis for practical design of heat pump ground grids. The intermittent type test appears to be much more closely related to the operation of an actual installation than the continuous run.

Little more can be added to the author's statements concerning the need for detailed comparison of results from various heat pump installations. The problem is indeed complex when the multiplicity of variables involved is reduced to a common plane and considered. Further comparison of various researches and actual operations, however, could yield an acceptable ground coil design method dependent upon soil properties and ambient conditions at each different site.

J. V. BORRY, Minneapolis Minn.: I was wondering, Professor, about your remark regarding the moisture content of the soil. In making this determination, did you use a control point so that you could separate the variations in soil moisture content caused by the climatic conditions from those due to normal moisture migration to the cold pipes in the ground?

E. M. MITTENDORF, Shipman, Va.: I would like to know what is the explanation for the difference between the Fig. 5, the 10-hour and 15-hour continuous run. There seems to be quite a difference there.

AUTHOR'S CLOSURE: Mr. Fluker has presented an excellent analysis of the possible effects of *on and off time* when operating heat pumps. In some reported tests, such as those by Penrod in Kentucky, the larger daily periods were broken up into 2 or more shorter periods. For the most part we made our daily runs as a single run for the following reasons. First, more attendance help would be required for multiple runs and such was not available; second, we desired to get good experimental values for the longer periods as well as for the short ones since during the colder spells it is the results obtained for the longer runs which must be considered for design purposes; third, the overall average results over a season or even a month would be approximately the same whether the daily runs were continuous or broken up into shorter multiple runs. The shorter multiple runs should show somewhat better overall results than single daily runs. The use of results from the latter type would afford some factor of safety in design.

Mr. Mittendorf questions the reason for the difference between the 10- and 15-hour runs in Fig. 5. During any continuous run the rate of heat flow to the tube decreases as the operating time is extended because the temperature of the surrounding soil approaches that of the tube. During the first few hours this change takes place rapidly but as infinite time is approached the rate of heat flow tends to approach a more nearly constant but much lower value. Because 10 hours is an operating time often used during actual intermittent operation and 15 hours would seldom be reached, the choice of these two gives some idea of the average used value and its relation to a limiting value.

In answer to Mr. Borry, the soil measurements were taken at a position far enough from the ground grid that moisture migration was not a factor, and thus the soil moisture values represent variations due to climatic conditions.

1585

No. 1585

INTERMITTENT HEATING AND COOLING OF BUILDINGS

By P. L. PFENNIGWERTH* AND MERL BAKER**, LEXINGTON, KY.

AN IMPORTANT reduction in the cost of heating or cooling intermittently occupied buildings may be achieved by decreasing or terminating the conditioning plant during the unoccupied period. Heating or cooling must be resumed prior to occupancy in order that the temperature be restored to a comfortable level. The time required for preconditioning depends upon the thermal capacity of the building walls and furnishings, the temperature drift of the structure, outside air temperature and the plant capacity.

The plant must have sufficient capacity to meet the most severe load and this service requires that the plant rating be greater than that required for the steady state design load. The excess capacity depends on the time allowed for preconditioning and the thermal capacity of the building. Temperature drifts during inactive periods may be limited by a set back control, thereby alleviating the load during the preconditioning period.

The thermal storage of the outside walls in massive structures limits change of the temperature gradient near the weather side of the wall and it is practically unaltered during an inactive period of several hours. The temperature gradient at the inside wall surface, however, diminishes with time, and will possibly reverse in sign if a significant glass area exists. The decreased temperature gradient at the inside wall surface represents a reduction in heat exchange between the walls and room air during cooling periods. The room air receives heat from inside partition walls and furnishings and loses it through exposed glass, outside walls, and floors or ceilings.

To determine the temperature to which a structure will drift in a given time requires the application of transient heat flow analysis. Because of the complexity of heat transfer through the outside building walls, the application of theoretical equations is usually not feasible. The method of finite differences offers a practical

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and accurate means of solution. The heat transfer through glass and thin walls may be treated as steady state problems; the heat loss rate is proportional to the inside-outside temperature difference.

Boundary conditions are established enabling the application of this method to determine the building temperature drift during an inactive period, the time required for preconditioning, and the requisite plant capacity.

SOLUTIONS BY METHOD OF FINITE DIFFERENCES

The method of finite differences is well recognized as a means of analyzing complex transient heat flow problems. It is of academic interest only until it is sup-

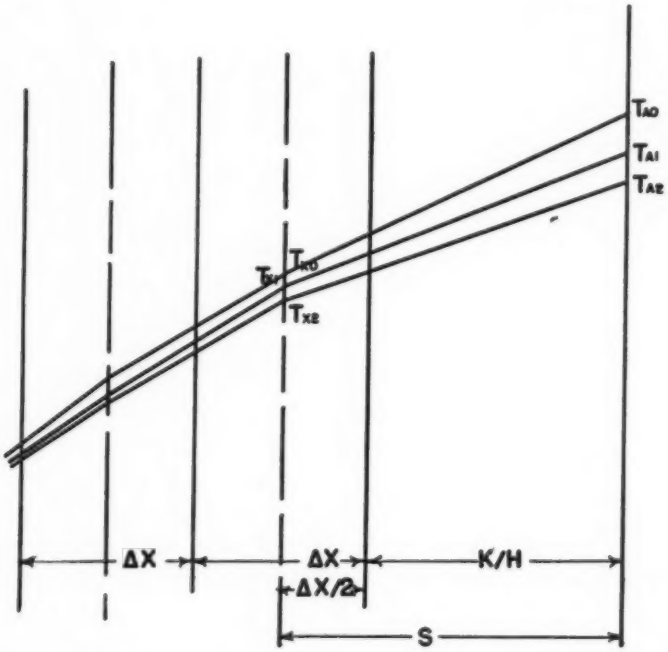

FIG. 1—LINE CONSTRUCTION DURING THE COOLING PERIOD

ported by boundary conditions prescribed by the particular problem under consideration. It is therefore the objective of this presentation to establish inside surface boundary conditions permitting analysis of intermittent heating and cooling problems and to discuss pilot experiments conducted for validation.

Fig. 1 depicts the construction technique pertinent to determination of the temperature drop that occurs after termination of the heat supply to a heated building. During this cooling period, heat is liberated from the room air, parti-

tion walls and furnishings equaling that transmitted through the bounding surfaces—external walls, roof, floor, and glass. Building walls are usually massive and must be treated in accordance with the laws of transient heat flow while floors, ceilings and windows can usually be analyzed by steady state relationships. Partition walls may be sufficiently massive to necessitate the use of a transient analysis, however, for simplification the heat liberated by partition walls may be approximated as follows:

$$Q_p = c_p m_p \Delta t n$$

where

Q_p = heat liberated from partition walls in Δt hours.

c_p = specific heat of partition walls.

m_p = mass of partition walls.

Δt = temperature change of room air in $\Delta\tau$ hours.

The temperature decrement, Δt, assumes that the wall temperature is uniform and follows that of the room air, a consideration which achieves expediency in application. The n term is introduced to compensate for the thermal lag of the partition wall and is defined as the ratio of the mean wall temperature decrement to the room air temperature decrement. This ratio may be approximated by calculation or experimental observation.

A mathematical expression relating the heat released from storage to that lost from the inside of the structure may be stated as follows:

$$\begin{aligned} 0.24 m_a(\Delta t_a) + n c_p m_p(\Delta t_a) + c_f m_f(\Delta t_a) \\ = \left\{ k_w A_w \left[\frac{\Delta t}{\Delta x}\right] + U_r A_r (t_a - t_r) \right. \\ \left. + U_b A_b (t_a - t_b) + U_g A_g (t_a - t_o) \right\} \Delta\tau \cdots\cdots (1) \end{aligned}$$

where

A = surface area, square feet.

C = specific heat Btu per pound (Fahrenheit degree).

k = thermal conductivity Btu per (hour) (foot) (Fahrenheit degree).

m = mass of material.

n = ratio of the mean change of the partition wall temperature to the air temperature change during $\Delta\tau$ hours.

U = overall heat transfer coefficient Btu per (hour) (square foot) (Fahrenheit degree).

Δt = change in temperature, Fahrenheit degrees in $\Delta\tau$ hours.
Subscripts a, b, f, g, o, p, r, and w refer to inside air, floor, furnishings, glass, outside air, partitions, roof or ceiling and outside walls respectively.

The method of finite differences is to be utilized in evaluating $\Delta t/\Delta x$. Fig. 1 depicts the temperature gradient at the surface of the inside wall and through the inside film. For the time interval $\Delta\tau$, the inside air temperature changes from $t_{a,0}$ to $t_{a,1}$ and the temperature within the wall at the center of the first Δx section changes from $t_{x,0}$ to $t_{x,1}$. Substituting for $\Delta t/\Delta x$ and considering the transmission

loss to be dependent on the average air temperature over the time increment, $\Delta\tau$, equation 1 becomes

$$0.24\, m_a (t_{a,0} - t_{a,1}) + nc_p m_p (t_{a,0} - t_{a,1}) + c_f m_f (t_{a,0} - t_{a,1}) = \left\{ k_w A_w \left[\frac{(t_{a,0} - t_{x,0}) + (t_{a,1} - t_{x,1})}{2S} \right] + U_r A_r \left[\frac{(t_{a,0} + t_{a,1})}{2} - t_r \right] + U_b A_b \left[\frac{(t_{a,0} + t_{a,1})}{2} - t_b \right] + U_g A_g \left[\frac{(t_{a,0} + t_{a,1})}{2} - t_o \right] \right\} \Delta\tau \quad \text{----------(2)}$$

where

s = distance between the equivalent inside film and center of first section, feet (see Fig. 1).

t_a, t_x, t_o, t_f, t_r = temperature of inside air, center of first wall section adjacent to inside air, outside air, beneath floor, and above the roof or ceiling, respectively, Fahrenheit.

Equation 2 is based on the average of the inside air temperature at the beginning and end of the time interval. Since the inside air temperature after a time lapse of $\Delta\tau$ hours is the object of calculation, it is expedient to approximate all transmission losses on the basis of t_a at the beginning of the interval. The value of $t_{a,1}$, end of interval, approximated on this basis may be employed to obtain an approximate mean for the interval, $(t_{a,0} + t_{a,1})/2$. The approximate mean temperature is used in a repeat calculation to determine $t_{a,1}$ more accurately. The successive approximations are continued until the desired degree of accuracy is achieved. For small time intervals, 2 approximations will usually suffice. By considering t_a the average inside air temperature over the time interval $\Delta\tau$, the statement of Equation 2 may be simplified.

$$\Delta t_a = \left[k_w A_w \left(\frac{\Delta t}{\Delta x} \right) + U_r A_r (t_a - t_r) + U_b A_b (t_a - t_b) + U_g A_g (t_a - t_o) \right] \Delta\tau \Big/ [0.24\, m_a + nc_p m_p + c_f m_f] \quad \text{-------(3)}$$

For a particular study, constant factors may be combined and Equation 3 expressed.

$$\Delta t_a = \left\{ \left[2SC_1 \left(\frac{\Delta t}{\Delta x} \right) + 2C_2 (t_a - t_r) + 2C_3 (t_a - t_b) + 2C_4 (t_a - t_o) \right] \Delta\tau \right\} \Big/ C_5 \quad \text{--------(4)}$$

where

$C_1 = k_w A_w / 2S$; $C_2 = U_r A_r / 2$;
$C_3 = U_b A_b / 2$; $C_4 = U_g A_g / 2$
$C_5 = 0.24\, m_a + n\, c_p\, m_p + c_f\, m_f$

After $t_{a,1}$ is located in Fig. 1, this point is connected with $t_{x,1}$, completing the temperature distribution curve after a time lapse of $\Delta\tau$ hours. The temperature t_{x2} is located in the manner depicted by Fig. A-2, Appendix A, and $t_{a,2}$ is computed by Equations 3 or 4 by advancing by one the subscripts referring to the inside air.

A similar development is followed during the study of the preconditioning period of the cycle. During the preheat period the rate of heat supply exceeds the building losses, and the temperature of the structure rises. The following relationships are established in accordance with the notation of Fig. 2. The energy supplied,

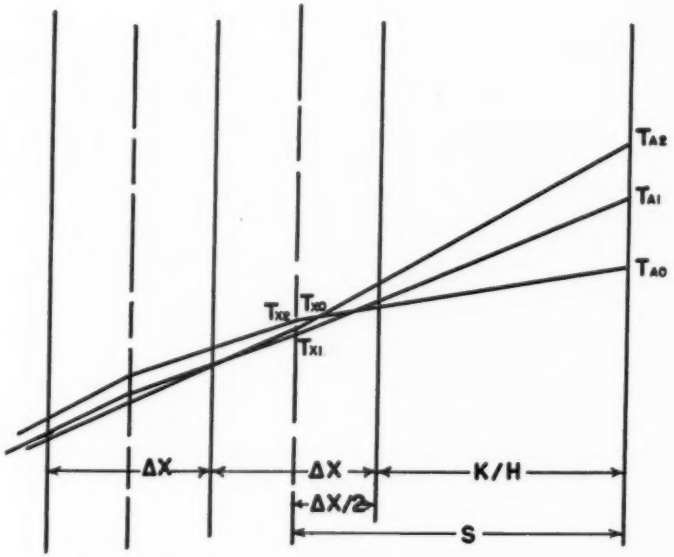

FIG. 2—LINE CONSTRUCTION DURING THE HEATING PERIOD

$R\Delta\tau$, is either transmitted through the boundaries of the structure or stored within the walls or internal masses: In equation form:

$$R\cdot\Delta\tau=(q_w+q_r+q_b+q_g)\Delta\tau + (0.24m_a+nc_pm_p+c_1m_f)(t_{a,1}-t_{a,0})\cdots(5)$$

where

R = heat supply rate, Btu per hour.

q = rate of heat loss, Btu per hour.
Subscripts w, r, b, and g previously defined.

Basing all losses on the average of the inside air temperature at the beginning and end of the time period $\Delta\tau$, Equation 5 may be written

$$\left\{R-K_w A_w\left[\frac{(t_{a,o}-t_{a,o})+(t_{a,1}-t_{a,1})}{2S}\right]\right.$$
$$-U_r A_r\left[\frac{(t_{a,o}+t_{a,1})}{2}-t_r\right]-U_b A_b\left[\frac{(t_{a,o}+t_{a,1})}{2}-t_b\right]$$
$$\left.-U_g A_g\left[\frac{(t_{a,o}+t_{a,1})}{2}-t_o\right]\right\}\Delta\tau$$
$$=(0.24\,m_a+nc_p m_p+c_f m_f)(t_{a,1}-t_{a,o}) \cdots (6)$$

Solving for $t_{a,1}$ gives

$$t_{a,1}=\{R\Delta\tau+[C_1(t_{s,o}+t_{a,1}-t_{a,o})$$
$$-C_2(t_{a,o}-2t_r)-C_3(t_{a,o}-2t_b)$$
$$-C_4(t_{a,o}-2t_o)]\Delta\tau+C_5 t_{a,o}\}$$
$$/\{C_5+[C_1+C_2+C_3+C_4]\Delta\tau\} \cdots\cdots (7)$$

Also by analogy with the formulation of Equation 4

$$\Delta t_a=\left\{R\Delta\tau-\left[2SC_1\left(\frac{\Delta t}{\Delta x}\right)\right.\right.$$
$$-2C_2(t_a-t_r)-2C_3(t_a-t_b)$$
$$\left.\left.-2C_4(t_a-t_o)\right]\Delta\tau\right\}\Big/C_5 \cdots\cdots\cdots (8)$$

The constants are the same as defined in conjunction with Equation 4. The time required for preheating is determined by the successive application of Equations 7 or 8 until the inside air temperature reaches the normal level.

Description of Experimental Apparatus

To verify the proposed equations, a test box was built to maintain controlled conditions. This box representing a building space or a room was constructed of 1 5/8 in. dry fir wood planks. The shape was approximately cubical and the inside surface area and inside volume were 35.94 sq ft and 17.53 cu ft respectively.

In order to approach the ideal case of unidirectional heat flow through the walls, the inside edges were aligned and the edge space was filled with granulated cork board insulation. The plank joints were butt glued to minimize discontinuities. Metal fasteners were used to assemble the 6 wall sections, but were placed at the edges only and morticed into the cork board.

The room furnishings were simulated by 2 beds of steel lathe turnings. One basket containing 25 lb of chips was supported on runners about 1 in. from the floor, and the second containing 35 lb of chips was supported at the approximate mid-plane of the box. Lathe turnings were used because of their large surface-to-mass ratio which would minimize temperature lag and thus closely assume temperature equality between the chip mass and air.

Heat was supplied by an electrical resistance heater having 10 helical nichrome coils extending the width of the box, and spaced uniformly over the length. The

heater and frame contributed about 2 lb to the mass of the furnishings, and was suspended in a horizontal plane, mid-way between the 2 chip beds. By virtue of this arrangement the radiation component of the heater was intercepted by the chip beds and heat transfer to the walls of the box was essentially by convection. The heater was designed for a surface temperature of about 600 F at an output of 770 watts in free air.

The outside environment of the box was maintained by placing it within a constant temperature cold room. The air in the room housing the box was recirculated through the evaporator coil by forced convection which insured thorough mixing of the environment air.

INSTRUMENTATION

All temperatures were measured with thermocouples in conjunction with both a precision type and a recording potentiometer. In order to measure the representative space temperatures, 9 thermocouples were distributed throughout the air volume and chip beds and could be read either singly or in parallel. The inside and outside surface temperatures were determined by 12 thermocouples held in the plane of the surface by strips of cellulose tape. To enable an additional check on calculated results, thermocouples were also embedded within the walls at intervals between the inside and outside faces. One set was placed at the geometric center of each of the 6 faces and at a plane 0.4 in. within the wall measured from the inside surface. These 6 thermocouples could be read either singly or in parallel. In addition, one wall contained additional thermocouples at the center plane and at a section 0.4 in. within the wall measured from the outside surface. In order to minimize disturbance to the temperature pattern within the walls, number 30-gage thermocouple wire was used, and the leads were pressed in the butt joint parallel to the plane of the wall surface for several inches from the junction point. The heater capacity was measured by a precision wattmeter and regulated by a variable output transformer. A photograph of the test box is shown in Fig. 3.

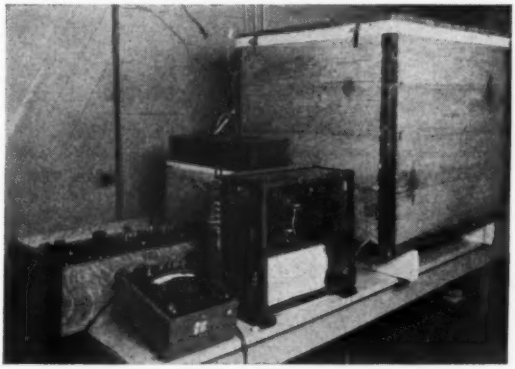

FIG. 3—EXPERIMENTAL APPARATUS

DETERMINATION OF PHYSICAL PROPERTIES

Auxiliary tests were conducted to determine as accurately as possible the density and specific heat of the wood. By weighing a given volume the density was found to be 27.8 lb per cu ft. The specific heat was determined as 0.37 Btu per (lb) (F deg) by the method of mixtures. The thermal conductivity was calculated using measured values of energy input to the text box at steady state conditions, wood thickness, and surface temperature differences. The average value was determined to be 0.712 Btu per (hr) (sq ft) (F deg).

DISCUSSION OF EXPERIMENTATION

Calculated and experimental results for the test box are compared by the curves of Fig. 4. Curve A represents the average of 2 nearly identical cool-down tests

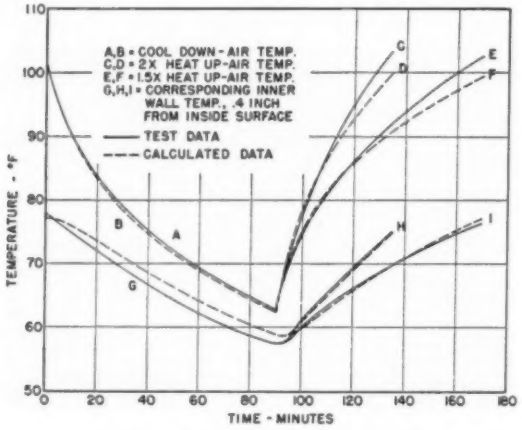

FIG. 4—COMPARISON OF CALCULATED AND EXPERIMENTAL STRUCTURE TEMPERATURE

conducted on different days, but under the same conditions. Curve B was obtained by application of Equation 1 which for all practical purposes is coincident with Curve A. Experimental values for the physical properties together with film coefficients of 1.49 inside and 2.52 outside, measured at the initial time, were employed in the determination of Curve B. Sample calculations are presented in Appendix B.

Curve C depicts test results for the preheat portion of the cycle during which the temperature was restored to the initial value. This occurred in 41 min, with the heat input being twice that required to maintain the desired inside temperature under steady state conditions.

Equation 5 was employed to locate points of Curve D. The ascending warm air currents over the central area of the box in conjunction with the descending currents near the cold wall apparently produced an important increase in the inside film coefficient over the value determined for steady state.

A mean value for these film coefficients during preheat was calculated based upon the rate of energy supplied to the box less the quantity stored in the chip bed and air mass during the same period. Using the mean measured temperature difference across the inside film during this period, the representative film coefficient was then calculated. As revealed by a comparison between Curves C and D the calculated temperatures are higher than corresponding test values during the early part of the period, but undergo a reversal, with the calculated temperature falling approximately 3 F deg below the experimental value at the final time. Although other factors may be pertinent, the deviation is believed to be caused by an actual film coefficient higher than the mean which was used for calculation, during the first part of the cycle and lower during the latter part. A similar comparison is observed between Curves E and F in which case the rate of preheating is 1.5 times the steady state value.

The temperature lag within the wall is evident from a study of Curves G, H, and I which represent values at 0.4 in. within the wall measured from the inside surface. The deviation between the calculated and experimental curves is believed to be insignificant.

APPLICATION

Calculations for an actual structure may be made in exactly the same manner as discussed in this article for the experimental box. It will be necessary to ascertain the thermal properties of the building material and make certain simplifying assumptions such as the relative temperature change of the furniture compared to the room air. If the furniture is well distributed and not massive, it may be assumed to approximate the room air temperature at all times.

CONCLUSIONS

The method of finite differences may satisfactorily be employed to predict the rate of temperature change and time required for preconditioning the intermittent heating or cooling of buildings. Previous reports have dealt with this application; however, the contribution of the present analysis is believed to be unique in the treatment of the boundary conditions.

The resulting equations are confirmed within experimental error by laboratory studies. The aberrations between calculated and observed results are believed to be primarily caused by the varying inside film coefficient during the experimental studies. An unstable and higher coefficient than that existing during steady state conditions is anticipated during the preheat portion of the cycle for the prototype structure.

Based on the boundary technique presented, the method of finite differences may be applied to any structure to evaluate any of the following: (*a*) Temperature drift during inactivity of the heating or cooling plant; (*b*) Time required for preconditioning; (*c*) Economy of intermittent operation; (*d*) Heat gain at the inside surface during isothermal cooling operations due to fluctuation of outside wall surface temperatures.

The following recommendations are believed to be important in achieving a more complete understanding of intermittent heating and cooling of buildings:

1. Investigation of the inside film coefficient during the preconditioning period of a full scale structure.
2. Evaluation of the rate of heat exchange between inside partitions and the surrounding air.

3. Analysis of possible heat savings from intermittent heating and cooling by application of the proposed equations to structures of various constructions and subject to various environments.

4. Analysis by the proposed boundary technique of the heat gain during summer cooling to a structure resulting from the intermittent heating and cooling at the weather side of walls.

APPENDIX A

Binder-Schmidt Method

The Binder-Schmidt method represents a solution of the basic heat flow equations based upon consideration of small finite differences. The result is an approximate

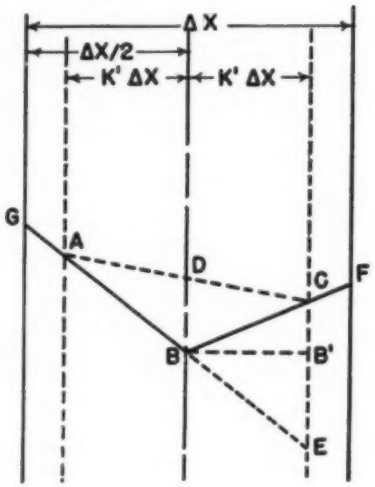

FIG. A-1—LINE CONSTRUCTION FOR THE METHOD OF FINITE DIFFERENCES

rather than exact solution. The approximate analysis, however, is so versatile as to be applicable to numerous engineering problems which are not solved feasibly by exact methods.

The principle of the Binder-Schmidt method is presented according to the analysis given by Raber and Hutchinson[1]. Consider the element of wall, Δx, Fig. A-1. Construction lines are located a distance $p = K\Delta x$ on either side of the center line of the section. Also consider that the temperature distribution through the section at some particular time is given by lines gb, bf.

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[1] Refrigeration and Air Conditioning, by B. F. Raber and F. W. Hutchinson (John Wiley & Sons, Inc., New York, N. Y.).

The rate of heat flow into the element from the left is

$$q_l = -kA\ (\Delta t/\Delta x) = -kA\ (\text{slope } ab) = +kA\ b'e/p$$

Heat leaves the section toward the right and is given by

$$-kA(\text{slope } bf) = -kA\ b'c/p$$

and that *entering* from the right is

$$q_r = kA(\text{slope } bf) = kA\ b'c/p$$

The total heat entering the section, Δx, is

$$(q_l + q_r)\Delta\tau = kA\ \Delta\tau(b'e + b'c/p)$$

and effects a mean temperature rise of

$$A\ \rho\ c\ (\Delta t)\ (\Delta x)$$

where

A = surface area normal to direction of heat flow, square feet.
c = specific heat of the wall material, Btu per (pound) (Fahrenheit degree).
k = thermal conductivity of the wall material, Btu per (foot) (hour) (Fahrenheit degree).
K = a factor defined by the equation $p = K\ \Delta\ x$.
p = distance of the 2 construction lines on either side of the section center line, feet.
Δx = the thickness of the section, feet.
$\Delta\tau$ = a time interval, hour.
Δt = a temperature difference, F.
ρ = density of the wall material, pounds per cubic foot.

Rearranging the terms of the previous equation and employing geometrical identities obtained from Fig. A-1.

$$(b'e + b'c)/p = [\rho\ c(\Delta t)(\Delta x)]/k\ (\Delta\tau) = ce/p$$

By similar triangles, $bd = \frac{1}{2}\ ce$, or $ce/p = 2\ (bd)/p$.
Substituting, then

$$[\rho\ c\ (\Delta t)\ (\Delta x)]/k\ (\Delta\tau) = 2\ (bd)/K\ (\Delta x)$$

By definition the thermal diffusivity, α, is equal to $k/\rho c$.
Solving for the distance bd, it follows:

$$b\ d = \Delta t\ \{[K/2\ \alpha]\ [(\Delta x)^2/(\Delta\tau)]\}$$

If,

$$(K/2\ \alpha)\ [(\Delta x)^2/(\Delta\tau)] = 1 \quad . \ . \ . \ . \ . \ . \ . \ . \ . \quad \text{(A-1)}$$

then, the distance bd represents the mean change in temperature of section Δx occurring during a time interval of $\Delta\tau$ hours.

This is significant because point d is located by the intersection of the section center line with a straight line connecting points a and c. Furthermore, Equation A-1 can always be made valid since Δx, $\Delta\tau$, and K may have arbitrary values for any particular wall. The section width must be taken such that an integral number of sections exists for the wall under consideration. The smaller the value of Δx, the greater is the accuracy of the solution. The recommended value of K ranges between $\frac{1}{4}$ and 1. By choosing both Δx and K, $\Delta\tau$ is established by Equation A-1.

Application of the Binder-Schmidt method requires that the initial temperature distribution through the wall be known. In most engineering problems, the temperature of the fluids adjacent to the wall surface is known rather than the temperature of the

surface. When such boundary conditions are known, a hypothetical thickness of the wall material is considered to replace the fluid film so that the thermal resistances are equivalent. The equivalent thickness of the film is k/h. Because of the negligible mass of material, thermal lag within the film is considered zero; and the slope of the temperature distribution line passing through the hypothetical film thickness and the adjacent solid wall surface must be identical.

The technique of applying the Binder-Schmidt method is illustrated in Fig. A-2 Curve O, a, c, a_2, c_2, a_3, c_3, i represents the initial temperature distribution through the material. According to the previous discussion, point d_1 is located by the intersection of a straight line connecting points a_1 and c_1 with the center line of the section. Points d_2 and d_3 are located similarly. The curve connecting points O, d_1, d_2, d_3, i represents the temperature distribution through the wall after a time lapse of $\Delta\tau$ hours.

Particular Application: Analysis of an intermittent heating problem requires consideration of the cooling period in addition to the preheating portion of the cycle. Fig. A-3

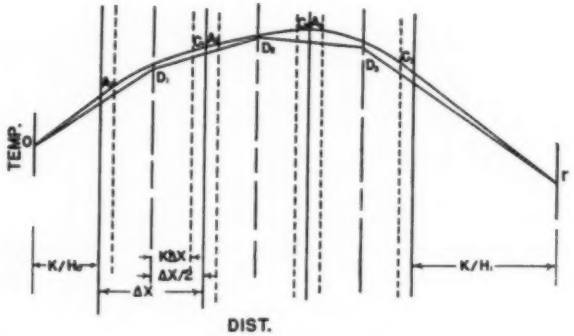


FIG. A-2—TECHNIQUE FOR TREATING SURFACE FILM COEFFICIENTS

illustrates the application of the Binder-Schmidt method to the cooling of a space after cessation of heat supply. The top curve represents the initial temperature distribution. For design purposes it is assumed that this represents a steady-state condition and therefore, the gradient is linear. Subsequent lines represent successive temperature distributions after intervals of $\Delta\tau$ hours, as determined from Equation A-1. The corresponding air temperatures after each time interval are calculated from Equations 3 or 4. This process is continued until the desired degree of cooling has been achieved.

The gradient existing at the end of this cooling period is then taken as the initial condition for the construction shown in Fig. A-4 which is the preheat period. Construction is continued, and air temperatures are calculated using Equations 7 or 8. This procedure is continued for an assumed heat supply rate, R, until the normal air temperature is achieved, and the preheat time is thus determined.

The particular constructions presented here pertain to the experimental test box. The preheat curve is for a heat supply rate 2.0 times the steady-state heat loss. The construction for a heat supply rate of 1.5 is identical except that because of the slower rate of heat up, construction lines are more closely spaced, and more time intervals are required to achieve the same normal air temperature.

APPENDIX B

Sample Calculations

Calculation of time interval: This is as follows.

ρ = 27.8 lb per (cu ft).
k = 0.712 Btu-inch per (sq ft) (hr) (F deg).
c = 0.37 B per (lb) (F deg).
Δx = ¼ × 1⅝ in. = 0.0331 ft (four wall sections).
α = k/ρc = 0.00577 sq ft per (hr).

From Equation A-1

$$\Delta\tau = k\,(\Delta x)^2/2\alpha = 0.02475 \text{ hr} = 2.85 \text{ min}$$

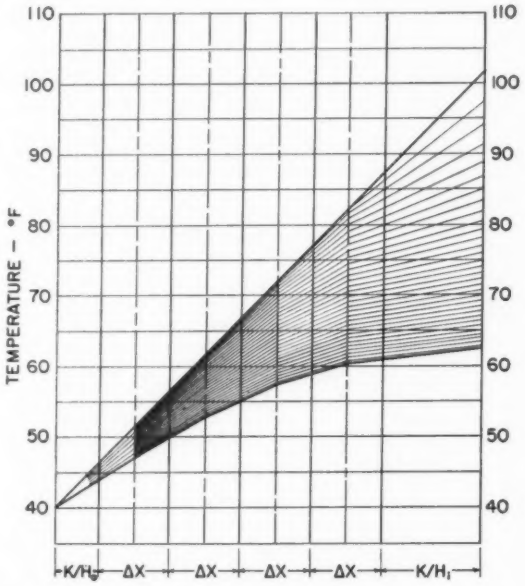


FIG. A-3—TEMPERATURE DISTRIBUTION THROUGH STRUCTURE WALL DURING COOLING

Cooling heat balance equation: Certain terms of Equations 3 and 4 are not pertinent to this application. Thus C_2, C_3, C_4 = O and also $m\rho = O$. The remaining terms are

k_w = 0.712 Btu-in. per (sq ft) (hr) (F deg)
A_w = 35.94 sq ft
$\Delta\tau$ = 0.0475 hr
m_a = 1.25 lb
c_t = 0.1092 Btu per (lb) (F deg)
m_t = 57 lb

Substituting these values into Equation 4

$$\Delta t_n = 0.0155\ (\Delta t/\Delta x) \quad \ldots \quad \text{(B-1)}$$

Sample calculation for Δt_n after time interval No. 4: From Fig. A-3

$(\Delta t/\Delta x)_3 = 170$ F per ft

Applying Equation B-1

$$\Delta t_4 = 2.63 \text{ F}$$
$$t_4 = 91.28 - 2.63 = 88.65 \text{ F}$$

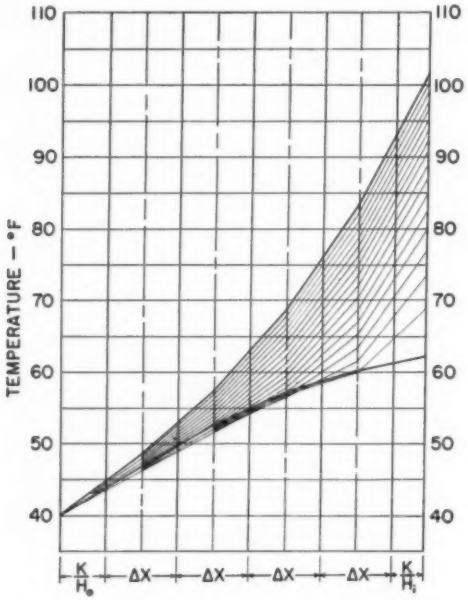


FIG. A-4—TEMPERATURE DISTRIBUTION THROUGH STRUCTURE WALL DURING PREHEATING

The construction of Fig. A-3 is then completed using this tentative value. Then $(\Delta t/\Delta x)_4 = 141$; and $(\Delta t/\Delta x)_{mean} = 155$ F per ft

Using this mean value in Equation B-1

$$\Delta t_4 = 2.40 \text{ F}$$
$$t_4 = 88.85 \text{ F}$$

The final construction is then made using this value. This method of approximation is only necessary during the initial periods when the rate of temperature change is high. In this instance, the 2 approximations differed by less than 0.1 deg after the eighth time interval, and only one calculation was therefore made for each of the remaining ones.

Preheat heat balance equation

$$R = 404 \text{ watts} = 1380 \text{ Btu per hr}$$

From Equation 8

$$\Delta t_a = 10.03 - 0.0155\, \Delta t/\Delta x \quad \text{(B-2)}$$

The method of application follows that for the cooling period.

DISCUSSION

C. S. LEOPOLD, Philadelphia, Pa., (WRITTEN): The authors have added the words *and cooling* to a paper dealing primarily with heating.

The engineering factors which must be considered in a cooling problem differ greatly from those which must be considered in a problem of heating.

Heating systems are customarily designed for an unoccupied building at night without illumination; the cooling function of air conditioning for an occupied building during the day with sun and artificial illumination.

My comments are directed principally to the application of this method of analysis to the cooling function.

The authors have not mentioned the effect of sun. If a plant is shut down at approximately 5 p.m., the solar gains of the western exposure add appreciably to the thermal storage.

The paper states that the plant rating be greater than that required for the steady state design load.

For the cooling function the plant need not be larger than for the ordinary peak during occupancy, as the low morning wet-bulb provides spare refrigeration capacity and the absence of artificial light prior to occupancy provides the spare for sensible cooling.

The paper states that floors and ceilings can usually be analyzed by steady state relationships. This would not be necessarily true for open floors in office buildings or for department stores. Frequently the thermal mass of floors and ceilings is comparable to the outside wall, and the ratio of wall to floor is relatively small.

An embedded thermometer on the second floor of a large department store showed only 1 deg change after 10 hours of pre-cooling but a woefully deficient plant was appreciably aided. I believe this was in part due to the changed temperature gradient in the floor and ceiling and in part to the cooling of the store fixtures.

For maximum advantage of pre-cooling, the setting of the central control stat should be lowered in order to sub-cool the air, and then be moved back to proper control point shortly before occupants arrive. In this way, a favorable temperature gradient may be set up in the structure even though the air temperature is optimum. This effect was quite noticeable in experiments with an hydraulic analogue.

S. F. GILMAN, Syracuse, N. Y.: This is an interesting application of the Schmidt method of transient heat flow analysis. It has the advantage that rather complicated boundary conditions can be handled, and the disadvantage that the amount of labor involved is frequently large. In our work we have overcome the disadvantage to some extent by using computing machines rather than graphical plots. This is ordinarily easily done because the temperature in one slice of the wall is the numerical average of the temperatures on either side during the preceding time increment.

In cooling load problems, we are dealing with cyclic variations of outdoor temperature and radiation. With a time increment of 2 minutes, a one-day cycle requires 720 complete calculations. Also, the initial temperature distribution is not linear, so that the convenient assumption used in this paper is not applicable. Moreover, for the periodic case, the final temperature distribution is usually not the same as the initially assumed distribution; hence, the solution has to be repeated with a different initial

distribution until there is agreement. As a consequence, it becomes expensive and time consuming. The method proposed by Nottage and Parmelee in recent papers is far quicker but is not so adept at handling the more complicated boundary conditions.

I am wondering why the authors selected a configuration with such little mass for checking the method. Fig. 3 indicates that after 30 minutes the temperatures are decreasing almost linearly and the heat flow is essentially constant. It is suggested that a heavier wall section, e.g. 8 inches of brick, be investigated. This and other laboratory studies and analyses would be essential before extending the method to the 4 items listed in the Conclusions. The difficulty is not in the Schmidt method, which has been validated by many investigators, but rather in the lack of information concerning such items as film coefficients, effect of radiation, and the physical properties of materials.

H. T. KUCERA, Western Springs, Ill.: Back in 1926, I believe it was, a committee of the *National District Heating Association* in co-operation with a committee of *The National Association of Building Owners and Managers* made extensive studies of the heating up and cooling down of structures. They made actual tests and readings in occupied buildings in several cities of the country. The results were compiled in a report with curves showing the heating up rate and the cooling down rate of the actual structures under practical operating conditions. The information contained in that report should be helpful in this study for comparison purposes.

AUTHORS' CLOSURE (Professor Baker): I think the discussions have raised very important points. Concerning Mr. Leopold's comment, I believe that even though our particular assumptions do not include all cooling problems, still the method might be applied to cooling and be of considerable help. We did not look into this cooling angle, but it would be very interesting to have someone do so.

A low capacity within the structure was used in order to get the greatest possible relative change within the walls. We felt that a better check on the method would be obtained with low thermal storage within the structure. While we realize that the thermal storage is low, we also feel that for actual structures with a higher ratio of internal storage to wall storage, the difference between the calculated and theoretical curves would be reduced.

We may or may not be wrong in that thought, but looking at the equations, it will be seen that with a high internal mass, the air temperatures will drop less rapidly, and extreme conditions would not be found on the surface. From the standpoint of confirmation, this was a strong point.

Periodic application of this method is very time consuming, but judging from a recent paper, it would appear that an analog computer should be able to work the problem out. With such a computer in existence, it would appear that all the engineer will have to do is to justify his assumption and then turn the work over to the computer to get the answer.

1586

No. 1586

PERFORMANCE OF ONE-ROW TUBE COILS WITH THIN-PLATE FINS, LOW VELOCITY FORCED CONVECTION

By D. G. SHEPHERD*, ITHACA, N. Y.

HEAT exchange matrices of tubes with continuous plate fins of small thickness are commonly used in refrigeration and air-conditioning systems with air flowing over the fins. The matrix or coil may have more than one row of tubes, but the fins are formed from a single sheet fitted over all the tubes. A matrix with a single tube row is shown in Fig. 1. Data exist for many types of finned surface, but the majority are of limited scope with respect to the many variables of tube and fin geometry. This investigation was undertaken to explore systematically the effects of tube size, spacing and location and the effects of fin size, spacing and thickness.

The results reported are confined to the characteristics of coils with a single row of tubes. Investigations are in progress on two-row coils and further work is envisaged on the effect of corrugations on one row plate-finned coils.

The coils were constructed in commercial fashion and dimensional tolerances were not restricted beyond normal industrial practice. Some of the small differences which occurred will be noted later, but none seem of sufficient magnitude to throw doubt on the overall validity of the results. Thus the results are directly applicable to the normal use of this particular type of coil.

METHOD OF TEST

The heat transfer characteristics were obtained by flowing air at room temperature over the finned side of the coils and circulating water at 150 to 170 F through the tubes. The rate of heat transfer was found from measurements of air flow and temperature rise and of water flow and temperature drop, the heat balance obtained in this manner being checked before any data were used. The unit conductance on the air side was obtained from the overall conductance by means of the *Wilson Plot*.[1,2,3] The essence of this method is that since the overall resistance to heat transfer is equal to the sum of the individual resistances in series, by holding

* Professor, Thermal Engineering, Sibley School of Mechanical Engineering, Cornell University.

[1] Exponent numerals refer to References.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956.

all resistances but one constant and varying that one, a line can be drawn which if extrapolated to a zero value of the variable resistance gives an intercept which is the sum of the other resistances. In this case there are 3 resistances in series, the air-side resistance, R_a, the water-side resistance, R_w, and the resistance of the tube wall, R_t. The last named is negligible, typical values being of the order of $R_a = 0.1$, $R_w = 0.001$, and $R_t = 0.0001$, all values in (hours) (square feet) (Fahrenheit degrees) per Btu, and thus a plot of the overall resistance R_0 against R_w yields R_a as intercept.

A closer analysis (see Appendix A) shows that if R is taken as reciprocal of h, the unit conductance, that is exclusive of the associated area, then

$$R_0 = (A_a/A_w)\, R_w + R_a \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

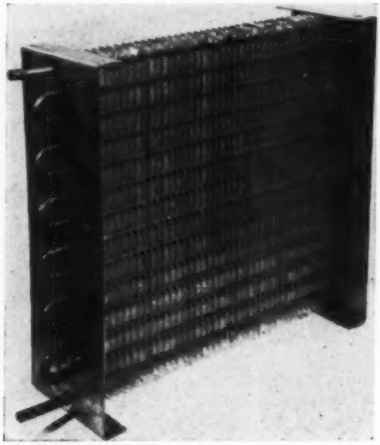


FIG. 1—TYPICAL COIL

Equation 1 shows that a plot of R_0 *vs* R_w should have a constant slope equal to A_a/A_w. Thus when one line has been determined, other lines with different air flow rates can be determined with relatively few points, since the slope of the line is known. This should also be true of different coils, since the resistance on the water side is unchanged in nature by a variation of the air side arrangement. If a relation for R_w is known exactly, then the slope is known initially, but it is best to plot R_w in terms of the variables without including the constant, as this is the largest source of uncertainty. Thus originally in this investigation, R_w was calculated from the well-known expression (Reference 2, page 168):

$$hd/k = 0.023\, (dG/\mu)^{0.8}\, (c\mu/k)^{0.4} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Use of this expression did not result in the same slope as given by the experimental observations, and thus R_w was plotted in terms of the water velocity, tube diameter and a temperature term embracing the variation of fluid properties, c, μ, k, and ρ. Equation 2 is recommended only to give values within ± 20 per-

cent and therefore in spite of its wide utility, complete agreement is not to be expected.

Equation 2 can be put into a simple form because the fluid properties can be accounted for by a linear temperature term for the narrow range of these tests in which the mean water temperature is between 125 and 160 F. The final form was

$$R_w \propto \frac{d_i^{1.8}}{m_w^{0.8}\,(1 + 0.01\,t_{wm})} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

t_{wm} = arithmetic mean water temperature, Fahrenheit.

The great majority of tests showed that the slopes of the R_0 *vs* R_w lines agreed with the predicted slope from the surface-area ratio, but there were exceptions discussed later.

The method of testing was then as follows: After thermal equilibrium was obtained, readings were taken at constant air and water flow rates. If a heat balance was obtained within 5 percent, then with the same air flow, the water flow was changed to a new setting and readings taken when equilibrium was obtained. The number of different water flow rates at each value of air flow varied, usually being a minimum of 3 but often greater if necessary to confirm the line with the calculated slope. The majority of coils could be grouped in threes, differing only in fin spacing. For each new group, sufficient water points were taken on the initial run to determine whether the theoretical slope was the correct one; in no case was the theoretical slope used in conjunction with only one or two experimental points.

The range of air flow was from about 15 lb per (min) (sq ft) to 75 lb per (min) (sq ft), corresponding to approach velocities of 200 fpm to 1000 fpm, with the initial pressure and temperature being atmospheric. The water flow rate varied between about 1.2 and 6 fps, with the inlet temperature varying between 150 and 170 F.

The overall thermal resistance, R_0, was calculated from the rate of heat transfer, q, by means of the relation

$$R_0 = (A_a\,\theta_m)/60\,q_w \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The surface area was computed from the coil geometry as the sum of the flat surface of the fins and exposed tube surface. The area of the edges of the fins was neglected as the fin thickness was 0.008 to 0.009 in. and also the area of the fin collars at the tube attachment was not taken into account. Both these areas were very small.

The rate of heat transfer, q, was taken as q_w, obtained from the water side, as this should be the highest value if there were any transfer of heat other than to the flowing air. However, heat balances within the 5 percent allowance showed no particular trend on either side of zero and thus q_w remains an arbitrary selection.

Calculation of the overall mean temperature difference (MTD), θ_m, was the subject of much consideration. The coil can be visualized as a single tube in cross flow. The initial results were plotted using (1) a log MTD, (2) an arithmetic MTD, and (3) a log MTD based on the mean water temperature and the air inlet and outlet temperatures. The last named gave the least scatter of points in plots of h_a against m_a and was used thereafter. Thus θ_m is defined as

$$\theta_m = \frac{(t_{wm} - t_{a1}) - (t_{wm} - t_{a2})}{\ln \dfrac{t_{wm} - t_{a1}}{t_{wm} - t_{a2}}} = \frac{\Delta t_a}{\ln \dfrac{t_{wm} - t_{a1}}{t_{wm} - t_{a2}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

DESCRIPTION OF APPARATUS

Fig. 2 shows schematically the arrangement of the test equipment. Air was sucked through the test section by a centrifugal fan. Following the test coil was an exit section with plenum and metering nozzle. Following the nozzle was a length of duct connecting to the suction side of the fan. Air flow was controlled by a slide valve at the fan discharge.

The 3 kw and 6 kw heaters, Fig. 3, were each capable of ¼ to ½ output. The 3 kw heater was connected electrically with a variable transformer so that the heat input was capable of fine control. The water was circulated by a centrifugal pump through metering orifices to the bottom of the test coil, returning thence to a small weighing tank. Both inlet and outlet pipes from the weighing tank were free. The water flow rate was controlled by a bypass at the discharge side of the pump.

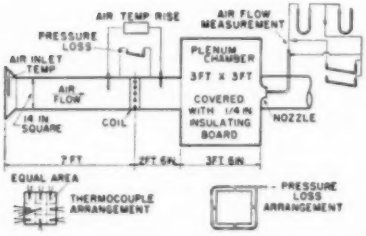


FIG. 2—SCHEMATIC ARRANGEMENT OF APPARATUS

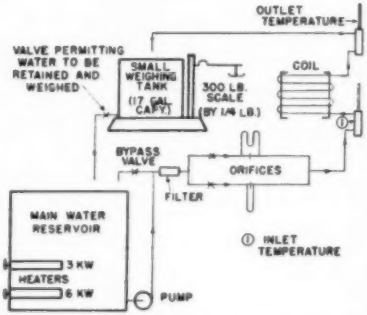


FIG. 3—SCHEMATIC ARRANGEMENT OF WATER CIRCUIT

The test coil and duct were insulated with approximately 1 in. thickness of hair felt for a distance of 1 ft either side of the coil.

INSTRUMENTATION

The measured quantities were air and water flow rates, air and water inlet and outlet temperatures and pressure drop across the test coil. Details of instrumentation appear in Appendix B.

DESCRIPTION OF TEST COILS

A schematic drawing of a coil is shown in Fig. 4, together with nomenclature used. The method of manufacture was to blank out the fins with a small collar or flange around the tube hole and to insert the separated tubes, which were expanded into the collars by forcing an oversize ball through them. The 180 deg bends were brazed on to form a continuous water passage. The bond between fin and tube was then as close as possible without being homogeneous. Tubes were copper and fins were aluminum and under the conditions of use there was little opportunity for corrosion. A re-test of one coil after being in storage many months showed no detectable change in performance. Only occasionally was it found on

inspection that a few fins were loose on the tubes and as these never exceeded 1 percent of the total number of fin-tube joints, it was concluded that the bonding was satisfactory in giving minimum thermal resistance next to a homogeneous joint.

In Table 1 is a list of the 38 coils tested. Test coils include some with fin depths of 1, 1½, and 2 in., tube pitches of ¾, ⅞, 1, 1⅜, and 2 in., and fins of 0, 3, 5, 7, 9, and 11 per in. All possible combinations of these arrangements were not tested. All coils were 14 in. square with a face area of 1.361 sq ft. All coils of the same width and nominal number of fins per inch did not have the same total number of fins due to manufacturing tolerances. The actual total number of fins was used in computing surface area. There is also a lack of true similarity in that the fin thickness of some coils was 0.008 in. as opposed to the majority which had a thickness of 0.009 in., again due to the manufacture of the coils being on a commercial rather than a laboratory basis. A comparison of 3 fin thicknesses on one particular arrangement was made directly and shown to be small.

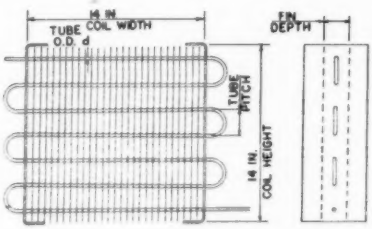


FIG. 4—COIL GEOMETRY

The thick outside supports of the coils are not included in the surface area in Table 1 as they normally amounted to only a fraction of the total surface area. However, the effect was tested in the case of one coil having the lowest total fin and tube area, by performing tests with and without the side walls insulated with ⅛ in. corkboard. The effect on R_0 was not detectable within the usual limits of scatter and as this was an extreme case, no account was taken of this area in computations.

PRESENTATION OF DATA

The number of separate tests in this phase of the investigation was of the order of 1000 and it is not possible to present here either the tabulated data or even each separate plot for a given coil.

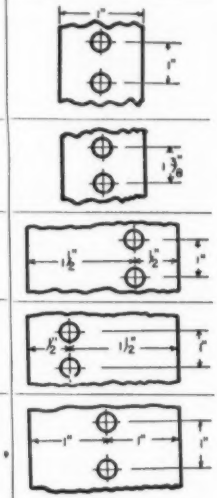
The following maximum ranges of test variables indicate the scope (exclusive of bare tube tests):

Water inlet temperature	t_{w1}	140 –178.2 F
Water outlet temperature	t_{w2}	101.3 –169.8 F
Water temperature drop	Δ_{tw}	3.5 – 69.6 F
Air inlet temperature	t_{a1}	67.2 – 90 F
Air outlet temperature	t_{a2}	73.65–127.4 F
Air temperature rise	Δ_{ta}	3.05– 49.4 F
Mean temperature difference	θ_m	29.4 – 90.6 F
Overall rate of heat transfer	q	40.4 –381 Btu per min

TABLE 1—DESCRIPTION OF THE 38 COILS TESTED

(All coils 14 x 14 in. overall, giving a face area of 1.361 sq ft)

Coil No.	Tube		Fin				Surface Area, Sq Ft				
	O. D. in.	Pitch, in.	Depth, in.	Thickness, in.	No. Per in.	Total No.	Tube Inside A_w	Tube Outside A_t	Fin A_f	Total A_a	
1	0.390	1.0	1.0	0.008	3	42	1.5	1.63	7.18	8.81	
2	0.390	1.0	1.0	0.008	5	72	1.5	1.60	12.33	13.93	
3	0.390	1.0	1.0	0.008	7	100	1.5	1.57	17.12	18.69	
4	0.390	1.0	1.0	0.008	9	128	1.5	1.55	21.9	23.45	
5	0.390	1.0	1.0	0.008	11	152	1.5	1.52	26.02	27.54	
6	0.390	1.0	..	..	0	..	1.5	1.67	..	1.67	
7	0.390	1.375	1.0	0.008	3	41	1.07	1.16	7.15	8.31	
8	0.390	1.375	1.0	0.008	7	99	1.07	1.12	17.27	18.39	
9	0.390	1.375	1.0	0.008	11	157	1.07	1.08	27.4	28.48	
10F	0.390	1.0	2.0	0.008	3	44	1.5	1.63	16.08	17.71	
11F	0.390	1.0	2.0	0.008	7	99	1.5	1.57	36.16	37.73	
12F	0.390	1.0	2.0	0.008	11	155	1.5	1.52	56.7	58.22	
10R	0.390	1.0	2.0	0.008	3	44	1.5	1.63	16.08	17.71	
11R	0.390	1.0	2.0	0.008	7	99	1.5	1.57	36.16	37.73	
12R	0.390	1.0	2.0	0.008	11	155	1.5	1.52	56.7	58.22	
13	0.390	1.0	2.0	0.009	3	41	1.5	1.63	14.99	16.62	
14	0.390	1.0	2.0	0.009	5	71	1.5	1.59	25.95	27.54	
15	0.390	1.0	2.0	0.009	7	99	1.5	1.56	36.2	37.76	
16	0.390	1.0	2.0	0.009	9	125	1.5	1.53	45.65	47.18	
17	0.390	1.0	2.0	0.009	11	155	1.5	1.50	56.7	58.2	



20	0.390	1.0	1.5	0.009	3	41	1.5	1.63	11.02	12.65	
21	0.390	1.0	1.5	0.009	7	99	1.5	1.56	26.6	28.16	
22	0.390	1.0	1.5	0.009	11	156	1.5	1.52	41.9	43.42	
28	0.322	0.875	1.0	0.009	3	41	1.30	1.54	7.23	8.77	
29	0.322	0.875	1.0	0.009	7	99	1.30	1.48	17.44	18.92	
30	0.322	0.875	1.0	0.009	11	157	1.30	1.42	27.66	29.08	
31	0.322	0.750	1.0	0.009	3	42	1.46	1.73	7.02	8.75	
32	0.322	0.750	1.0	0.009	7	101	1.46	1.66	16.89	18.55	
33	0.322	0.750	1.0	0.009	11	159	1.46	1.59	26.58	28.17	
34	0.390	2.0	1.0	0.009	11	157	0.75	0.75	28.68	29.43	
35	0.390	2.0	1.0	0.009	7	97	0.75	0.78	17.72	18.50	
36	0.390	2.0	1.0	0.009	3	41	0.75	0.81	7.49	8.30	
37	0.390	2.0	2.0	0.009	3	42	0.75	0.81	15.87	16.68	
38	0.390	2.0	2.0	0.009	7	98	0.75	0.77	37.0	37.77	
39	0.390	2.0	2.0	0.009	11	154	0.75	0.76	58.2	58.96	
40	0.390	1.0	1.0	0.009	3	41	1.5	1.63	7.02	8.65	
45	0.390	1.0	1.0	0.009	3	42	1.5	1.62	7.18	8.80	
46	0.390	1.0	1.0	0.011	3	42	1.5	1.61	7.18	8.79	

HEAT TRANSFER

Fig. 5 shows a typical plot of R_0 against the factor proportional to R_w. The bottom line of highest air flow shows the distribution of points obtained when a large number of values of R_w was taken to check the slope. The remaining air flows had fewer points, except the top line for lowest air flow in which additional points were taken, as the line of fixed slope through the first 2 points taken showed scatter with the third point. The line for 25.3 lb per sec air flow shows only two points and was taken for confirmation to make an extra point on the graph of h_a against m_a.

From plots similar to Fig. 5, the h_a value was found as the reciprocal of the intercept of the extrapolated line on the R_0 axis. These values of h_a were then plotted against the air mass flow m_a on logarithmic paper, resulting in plots of which Fig. 6 is typical.

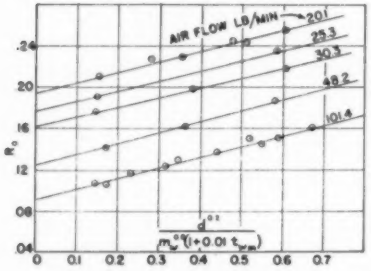


FIG. 5—TYPICAL PLOT OF OVERALL RESISTANCE *vs* WATER SIDE RESISTANCE FACTOR

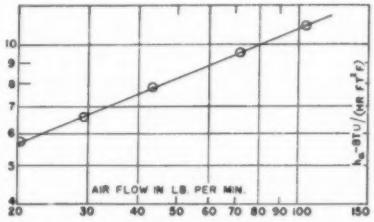


FIG. 6—TYPICAL PLOT OF OVERALL AIR SIDE UNIT CONDUCTANCE *vs* AIR FLOW RATE

These h_a *vs* m_a lines were then used to plot h_a against G_0, where $G_0 = m_a/A_0 = \rho_a V_0$, the specific mass flow in lb per (minute) (square foot of face area). Since all the coils had the same value of A_0 in this instance, the replot is only a shift of the abscissa, but G_0 is more basic than particular values of m_a. It might be noted that since $A_0 = 1.361$ sq ft and the average ambient density was about 0.0735, then $V \approx 10\ m_a$, where V_0 is the face velocity in feet per minute. Thus a value of m_a of 50 lb per min represents a velocity of about 500 fpm.

The h_a *vs* G_0 results are shown in Fig. 7, each diagram (A) to (J) being for a group of similar geometry except for fin spacing, while (K) shows the effect of fin thickness on one particular fixed arrangement having 3 fins per in. The lines are shown without experimental points for clarity.

PRESSURE LOSS

The pressure loss Δp_L was measured for all coils under isothermal conditions and for many coils during heating runs. A typical plot of Δp_L showing a comparison under heating and isothermal conditions is given in Fig. 8. It was found that the two could be correlated satisfactorily by using the density evaluated at the

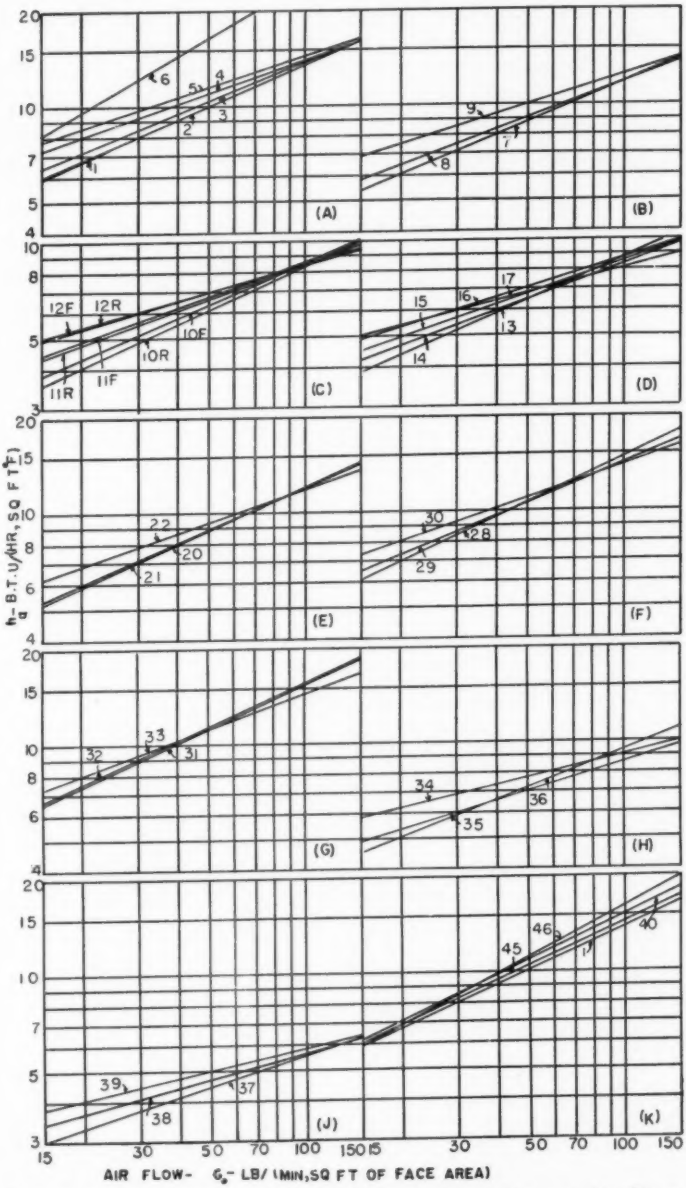


FIG. 7—OVERALL AIR SIDE UNIT CONDUCTANCE *vs* SPECIFIC AIR FLOW

mean air film temperature, taken as the arithmetic mean of absolute air and water bulk temperatures, that is

$$\Delta p_L \text{ (heating)}/\Delta p_L \text{ (isothermal)} = [t_{a1} + t_{a2} + t_{w1} + t_{w2} + 4(460)]/[4(t_{a1} + 460)]. \quad . \quad . \quad (6)$$

Values of Δp_L isothermal plotted against G_0 are shown in Fig. 9. Within the limits of scatter, which were very small, the lines are linear with the exception of a few for coils of small fin spacing where there is an upward trend at low flow rates.

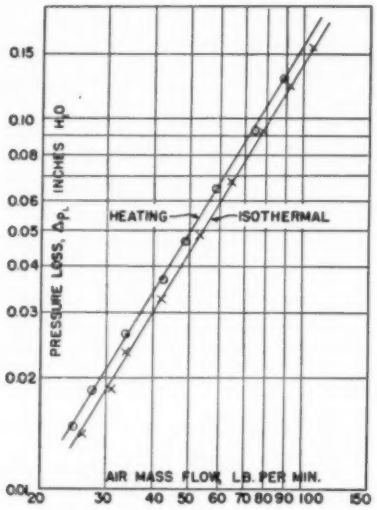


FIG. 8—PRESSURE LOSS *vs* AIR FLOW RATE—TYPICAL PLOTS OF ISOTHERMAL AND HEATING RUNS

This trend is similar to that for the air side coefficient of heat transfer. However, as the effect was very small in the range of common flow rates, the plots have all been taken as linear.

DISCUSSION OF RESULTS

From Fig. 7 it is seen that at low velocities, the combined air side conductance h_a is higher for the coils of small fin spacing and that as the air flow increases the lines tend to converge and sometimes cross over.

Fig. 7 (K) shows the effect of fin thickness on heat transfer. All coils were of 3 fins per in. and of the same geometrical arrangement, but No. 1 had a fin thickness of 0.008 in., Nos. 40 and 45 of 0.009 in. and No. 46 of 0.011 in. Coils 40 and 45 were nominally similar and are included to demonstrate the possible effect of departure from uniform conditions, because approximately 1/3 of the fins of coil 40

were bent in transit and were hand-straightened. The latter resulted in some large scale *waviness* and lack of parallelism of some vanes. The relatively similar performance of coils 40 and 45 demonstrates, in this instance at least, that coils of this type are not very sensitive to small departures from nominal conditions.

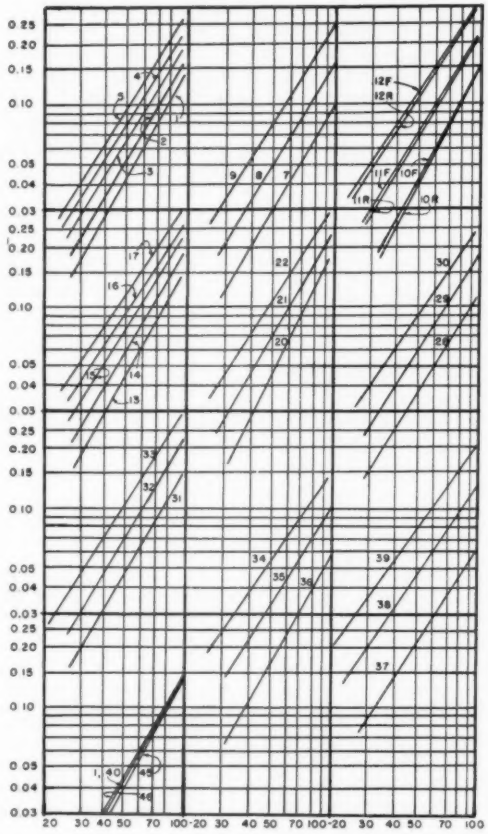


FIG. 9—PRESSURE LOSS *vs* SPECIFIC AIR FLOW RATE FOR ALL COILS

Fig. 10 shows the performance of coils similar in all respects except the position of the tube in the fin. Coils 13 to 17 were of the usual type with the tube symmetrically placed in the center of the 2-in. deep fin, while coils 10F to 12F had the tube ½ in. off center displaced towards the rear and coils 10R and 12R had the

tube displaced ½ in. towards the leading edge. It was thought that as the heat transfer coefficient from a tube in cross flow varies around the circumference, the altered temperature distribution in the fin, together with the altered air flow pattern, would result in a difference of combined conductance. However, Fig. 10 shows that the difference was slight and that, within the general limits of manufacturing and experimental tolerance, the eccentricity of the tube with respect to the fins exerts no discernible effect. Coils 10F to 12F and 10R to 12R are therefore not included in any subsequent analysis.

Attention is drawn to some inconsistent results. In particular the slopes of the R_0 lines for 3 groups of coils in plots similar to Fig. 5 did not conform to the values predicted on the area ratio basis. Thus coils 31 to 33 yield a slope about 35 percent greater than the theoretical coils, 34 to 36 about 10 percent less and coils 37 to 39 about 20 percent less. The experimental points gave consistent linear plots

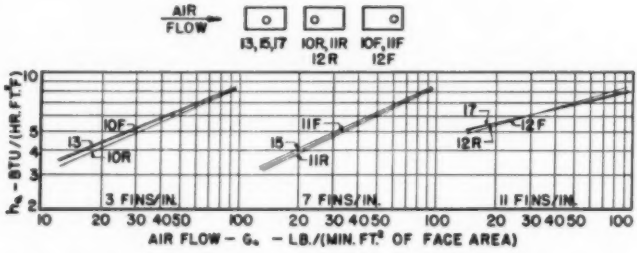


FIG. 10—COMPARISONS OF UNIT CONDUCTANCE FOR SYMMETRIC AND ECCENTRIC FINS

and obviously could not be forced into the nominal slope. In each group of 3 coils, the coils were consistent among themselves in following the area ratio requirement. One coil of each group was checked and rechecked, the re-tests involving reinstallation and collection of data by different operators. Observational and dimensional inspection showed no apparent differences in construction. The performance was therefore taken as representative, although it is regrettable that no constructional, installational, experimental, or theoretical hypotheses can be put forward to account for the apparently anomalous results.

Practical performance criteria are likely to be concerned, not with heat transfer coefficient alone, but with the overall rate of transfer per unit temperature difference and with the associated power loss due to pressure drop. Figs. 11, 12, and 13 show some of these overall performance criteria.

Fig. 11 shows q_0, the rate of heat transfer, Btu per (hour) (Fahrenheit degree) (square foot of face area), that is

$$q_0 = (h_a A_a)/A_0 = (h_a A_a)/1.361 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

The highest values of q_0 are for the coils having the most fins per inch, but an important conclusion from these results is that although there is an increase in q_0 for increase in fin depth from 1 to 1½ in. (coils 1 and 20, 3 and 21, 5 and 22), there is little or no increase in going from 1½ to 2 in. (coils 20 and 13, 21 and 15, 22

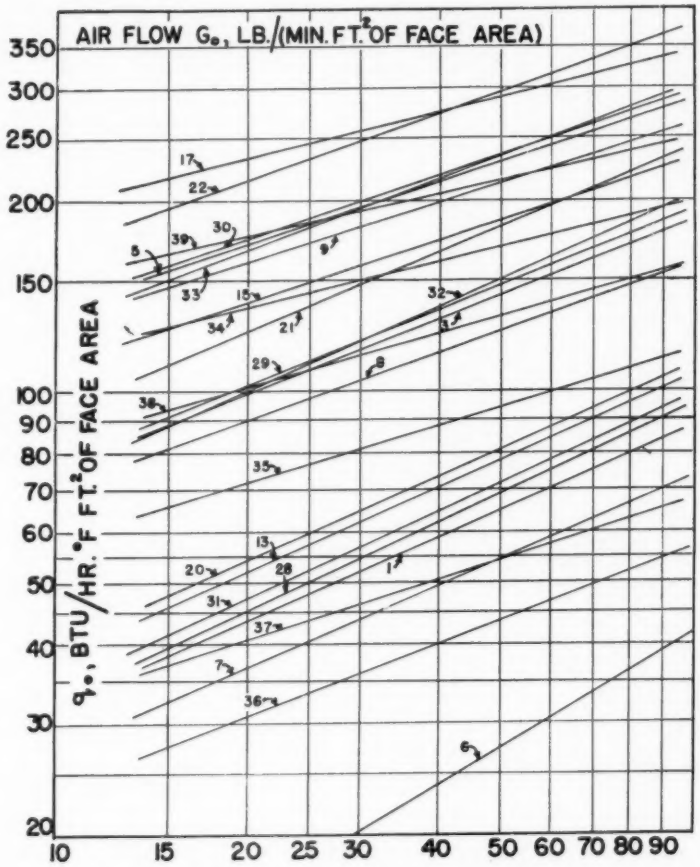


FIG. 11—COMPARISON OF OVERALL RATE OF HEAT TRANSFER FOR ALL COILS

and 17). The effect of tube spacing by itself is approximately proportional to that for film coefficient alone, as the surface area contributed by the tubes is small compared to the surface area of the fins.

Fig. 12 shows the ratio of the combined air side coefficient to the pressure loss, $h_a/\Delta p_L$. The coils of best performance with this criterion are those of wide fin spacing, wide tube spacing and shallow fins, the highest values being for coil No. 36 with one inch deep fins on 2-in. center, 3 fins per in. Any of the three methods of increasing the rate of heat transfer, number of fins, depth of fins, or number of tubes occurs only with proportionately greater increase of pressure loss. A better

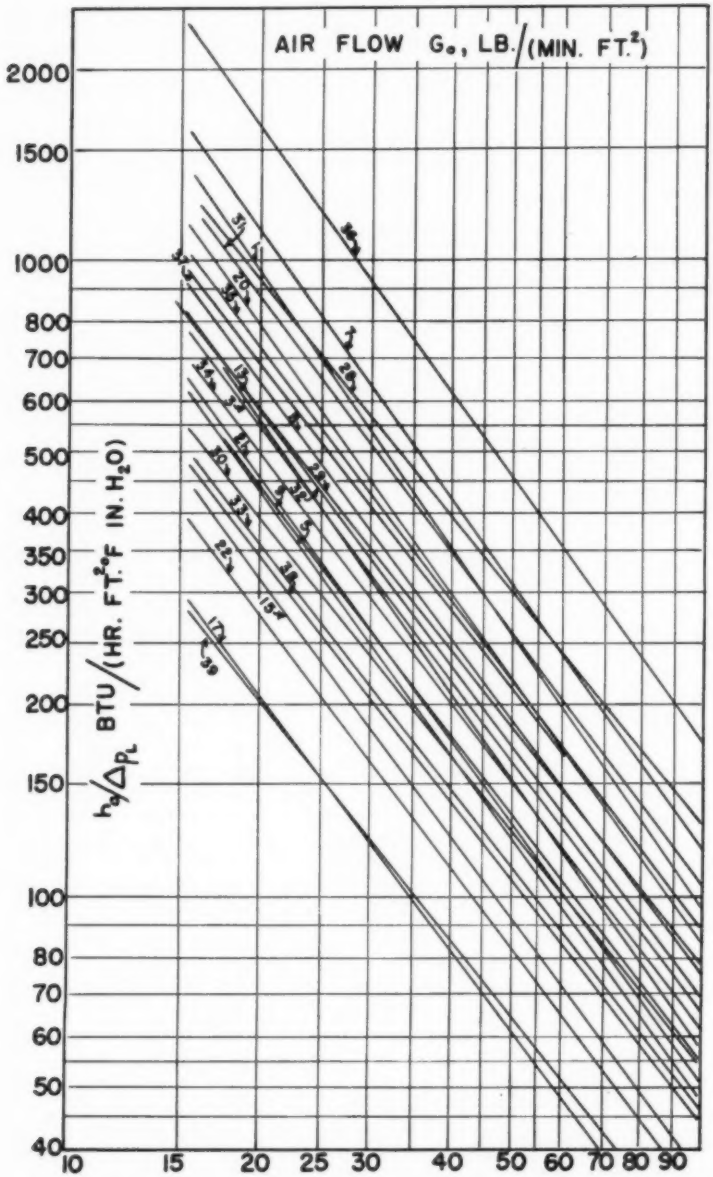


FIG. 12—COMPARISON OF UNIT CONDUCTANCE PER PRESSURE LOSS

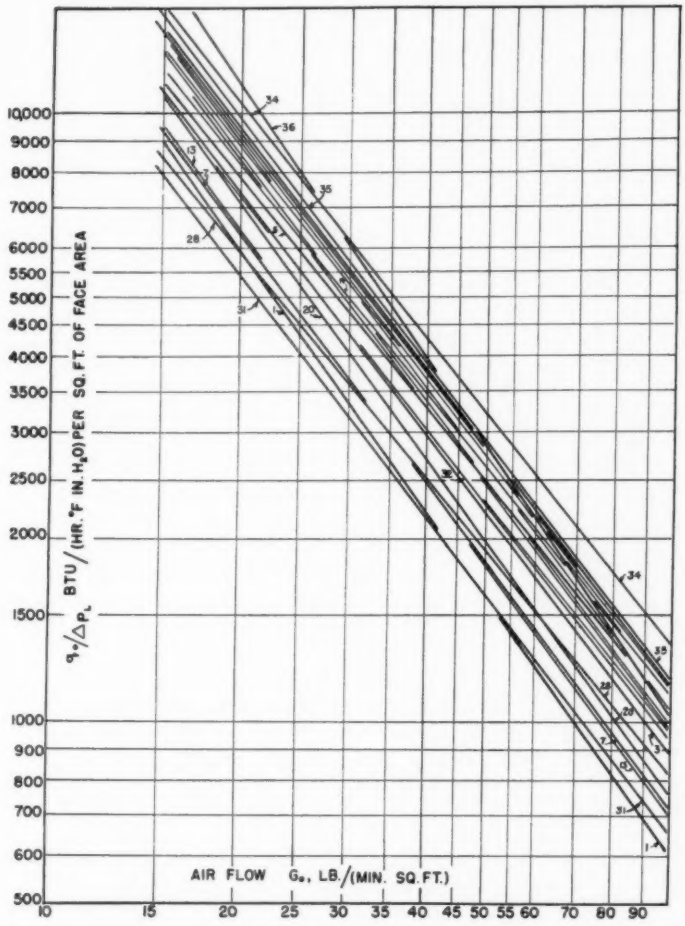


FIG. 13—COMPARISON OF RATE OF HEAT TRANSFER PER PRESSURE LOSS FOR ALL COILS

$h_a/\Delta p_L$ value is obtained by doubling the number of tubes rather than by doubling the fin depth and has approximately the same effect as doubling the number of fins.

Fig. 13 gives the ratio of rate of heat transfer per unit face area and unit temperature difference to pressure loss, $q_0/\Delta p_L$, which is perhaps the most significant design parameter if neither heat transfer nor pressure loss are by themselves limiting criteria. It will be seen that all coils now lie within a much narrower range than for either of the other two criteria, although this range is still about two to one. The best performance on this basis is achieved by coils of closely spaced fins of

small depth on wide tube spacing. Many of the coils have such a similar performance that they may be represented by one line.

The effects of fin spacing, fin depth and tube pitch alone are shown in Figs. 14, 15, and 16. Fig. 14 shows h_a, $h_a/\Delta p_L$, q_0, and $q_0/\Delta p_L$ plotted against the number of fins per inch for a constant air mass velocity, G_0, of 50 lb per (min) (sq ft) for

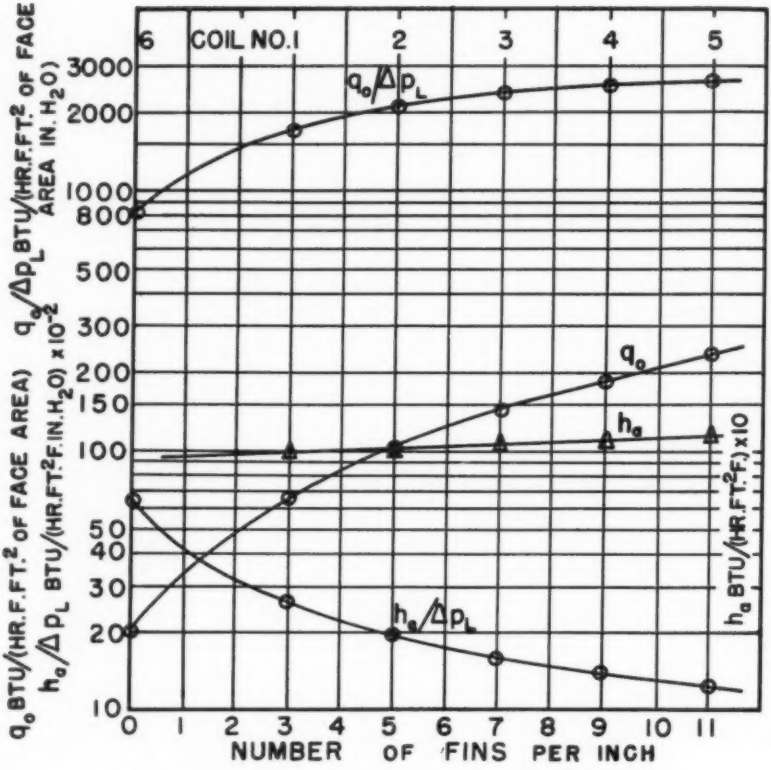


FIG. 14—EFFECT OF FIN SPACING ON COIL PERFORMANCE

coils 1–6 having a tube diameter of 0.390 in. with one inch spacing, coils 1 to 5 having 3, 5, 7, 9, and 11 fins per in., respectively, and coil 6 being bare. At this value of air flow (approx. 680 fpm), the effect of fin spacing on h_a is very small, but q_0 still increases continuously with the number of fins, and the maximum rate of heat transfer has not been reached with 11 fins per in., the largest number tested. However, the ratio of heat transfer to pressure loss, $q_0/\Delta p_L$, apparently reaches a maximum at about 7 fins per in., with little change thereafter.

Fig. 15 shows the various criteria plotted against the ratio of tube pitch to tube diameter for coils of one inch fin depth, 7 fins per inch, for G_0 = 50 lb per (min) (sq ft). Three of these coils, 3, 8, and 35, had a tube diameter of 0.390 in. and the other two (29 and 32) a tube diameter of 0.322 in. Within this small range of tube diameter, the pitch/diameter value appears a correlating parameter, with no effect due to actual tube diameter by itself. It is seen that increasing the pitch ratio reduces the values of heat transfer but increases the heat-transfer to pressure-loss ratios.

Fig. 16 shows the various criteria plotted against fin depth for coils of 0.390-inch tube diameter 1 in. pitch, 7 fins per in., for G_0 = 50 lb per (min) (sq ft) together

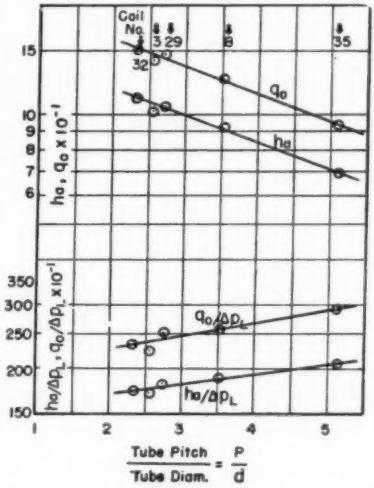


FIG. 15—EFFECT OF TUBE PITCH-TUBE DIAMETER RATIO ON COIL PERFORMANCE

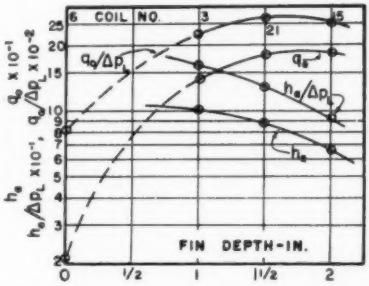


FIG. 16—EFFECT OF FIN DEPTH ON COIL PERFORMANCE

with the values for the bare tube coils, Nos. 3, 21, 15, and 6. This demonstrates clearly the decreasing value of fin depth above 1½ in.

ACKNOWLEDGMENT

The experimental results were obtained in a research project at Cornell University generously sponsored by The Salvatore Giordano Foundation, Inc., of Maspeth, New York. The author is indebted to members of the staff of the Department of Thermal Engineering for suggestions and criticisms and to several students who helped in obtaining the experimental data.

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NOMENCLATURE

English Letter Symbols

A_a = total air-side surface area, square feet.
A_f = fin air-side surface area, square feet.
A_t = tube air-side surface area, square feet.
A_w = tube water-side surface area, square feet.
A_0 = coil face area = coil width × height, square feet.
c = specific heat of water, Btu per (pound) (Fahrenheit degree).
d = tube outside diameter, inches.
d_i = tube inside diameter, inches.
$G_0 = m_a/A_0$ = air mass velocity based on A_0, pounds per (minute) (square foot of A_0).
h_a = combined air-side unit conductance, Btu per (hour) (square foot) (Fahrenheit degree).
h_w = water-side unit conductance, Btu per (hour) (square foot) (Fahrenheit degree).
k = thermal conductivity of water, Btu per (hour) (foot) (Fahrenheit degree).
m_a = air mass flow rate, pounds per minute.
m_w = water mass flow rate, pounds per second.
p = tube spacing or pitch, center-to-center, inches.
q = rate of heat transfer, Btu per minute.
q_0 = rate of heat transfer for unit face area per degree temperature difference = $q/A_0\theta_m$, Btu per (hour) (square foot) (Fahrenheit degree).
Q = rate of heat transfer, Btu per hour.
R_0 = overall unit resistance based on A_a, (hours) (square feet) (Fahrenheit degrees) per (Btu).
R_a = combined air-side unit resistance based on A_a, (hours) (square feet) (Fahrenheit degrees) per (Btu).
R_t = tube wall unit resistance, (hours) (square feet) (Fahrenheit degrees) per (Btu).
R_w = water-side unit resistance, (hours) (square feet) (Fahrenheit degrees) per (Btu).
t_a = air temperature, Fahrenheit.
t_w = water temperature, Fahrenheit.
U_0 = overall unit conductance based on A_a, Btu per (hour) (square foot of A_a) (Fahrenheit degree).
V = air velocity, feet per minute.

Greek Letter Symbols

Δp_L = overall pressure loss across coil, inches water.
θ_a = temperature difference between total air side surface and air, Fahrenheit.
θ_w = temperature difference between water and tube wall, Fahrenheit.
θ_m = mean temperature difference defined by Equation 5, Fahrenheit.
μ = fluid absolute viscosity, consistent units.
ρ = fluid density, pounds per cubic foot.

APPENDIX A

Equating the expressions for the rate of heat transfer (1) overall, (2) from the water, (3) through the tube walls, and (4) to the air,

$$Q = U_0 A_a \theta_m = h_w A_w \theta_w = \frac{(2\pi k \Delta t_t)}{ln \frac{d_0}{d_i}} = h_a A_a \theta_a$$

Neglecting the very small tube wall temperature drop Δt_t, then

$$\theta_m = \theta_w + \theta_a$$

whence

$$\frac{1}{U_0 A_a} = \frac{1}{h_w A_w} + \frac{1}{h_a A_a}$$

Multiplying through by A_a and using resistances as the reciprocal of conductances,

$$R_0 = \left(\frac{A_a}{A_w}\right) R_w + R_a, \text{ which is Equation (1).}$$

APPENDIX B

Air Flow: Air flow was measured by a standard nozzle in conjunction with a standard pitot-static tube, 2-, 4-, and 6-in. nozzle sizes being used to obtain accurate readings over the range. The size of the plenum chamber and the low velocities into it gave excellent entrance conditions to the nozzle. The static pressure (sub-atmospheric) and nozzle differential were measured on either a draft gage of 1-½-in. water capacity or on a U-tube manometer, both containing standard manometer oil. Air temperature at the nozzle was measured initially and found to be within 1 or 2 deg of the temperature immediately after the test coil, owing to the generally small air temperature rise and the insulation of exit duct and plenum chamber. For calculating air flow rate, the temperature was then taken as 2 F less than the measured air discharge from the coil. An error of 2 F in this measurement involves less than 0.2 percent error in air flow rate. The air flow was steady and once set by the outlet side valve remained constant indefinitely.

Water Flow: Water flow rate was measured by timing the collection of a known quantity through an interval determined by successive tipping of the beam. Once set, the water flow rate was steady over a long period, repeatable readings being obtained over a period of 3 min within an accuracy of ½ percent. The orifices in the line were uncalibrated and served only to set the flow approximately at a predetermined rate. The water circuit is shown schematically in Fig. 3.

Air Temperatures: Inlet air temperature was measured by a mercury thermometer reading to 1 F suspended with the bulb near the center of the intake. An accurate traverse of the intake by thermocouple showed that there was at times a temperature gradient of about 2 F, low at the bottom and higher at the top. The bulb was positioned in the region of mean temperature. Comparison of several thermocouple traverses with the thermometer temperature showed that the deviation was less than 0.25 deg F and as the air temperature rise was measured differentially, this deviation represented only a small error.

Temperature rise across the coil was measured by a thermopile consisting of 2 sets of 9 copper-constantan thermocouples connected differentially, each couple being placed approximately at the center of equal areas of the duct. The planes of measurement were 15 in. upstream and 12 in. downstream from the center line of the coil. The couples were of 24-gage wire, welded to form a bead and held in a ceramic tube supported in the duct wall. The thermocouple bead and exposed wires were covered with aluminum foil to minimize radiation errors. The emf was measured by a potentiometer on which the smallest divisions were 0.01 mv and it was possible to estimate the sub-divisions to 1/5 of a whole division.

The thermocouples were connected to a switch box so that not only the mean differential temperature could be read, but also the mean of the inlet and outlet temperatures and any one of the 18 separate temperatures alone. At one stage a completely separate duplicate differential thermopile was installed with the beads located about ½ in. from the initial set. Comparative readings showed that the mean temperatures

recorded were the same within negligible error. Exploration of each separate temperature in turn showed a varying temperature of up to 8 deg F at outlet, as might be expected, because although there was a 12-in. mixing length, the exact location of a bead with respect to a tube of the coil affects the local temperature. However, readings taken by moving the thermocouples slightly, in conjunction with the complete duplicate test previously mentioned, led to the conclusion that a satisfactory mean differential temperature was being obtained.

Water Temperatures: Water inlet and outlet temperatures were measured at points about 18 in. from the actual inlet and outlet tubes of the coil by means of mercury thermometers graduated to 0.2 deg F. The bulbs of the thermometers were directly immersed in the water in a mixing device. The temperatures were read to 0.1 deg F and with the high water velocity and direct immersion, very little thermal lag was observed.

Pressure Loss: Pressure loss across the coil was measured by static piezometer rings connected differentially to a draft gage. Each piezometer ring consisted of four $\frac{1}{64}$-in. holes, one in the center of each wall of the duct, 9 in. downstream of the coil. The very low values of pressure loss might have merited the use of a precision manometer, but in the first place the plotted results gave lines with relatively small scatter, thus yielding satisfactory comparisons, and in the second place, the measurement of loss by wall static holes does not necessarily give the true pressure loss. A precise determination of pressure loss would have involved a detailed traverse of the flow area with a pitot-static tube, and the labor involved in the many determinations required was not thought to be justified, in view of the fact that applications of this type of coil would be concerned only with static pressure measurements of the type used.

DISCUSSION

E. L. SARTAIN, Racine, Wis., (WRITTEN): The author of this paper has contributed a valuable piece of work relating coil geometry to heating performance. It is apparent that considerable time and effort were required to collect data, analyze results and work up the data into its final form.

The results of this study can be extended and readily applied to product design. For instance, it would be interesting to calculate air horsepower from the pressure drop and air weight flow rate. Air horsepower and heating capacity could then be plotted against air weight flow rate for each significant coil geometry. With this information, a suitable power unit could be selected for any design criterion.

The *Wilson Plot* employed by the author to determine the air side heat transfer coefficient for each coil is a very valuable tool, and this paper points out its applicability. Equation 3 shows that the water side thermal resistance is directly proportional to $d_i^{1.8}$. Fig. 5 shows a plot of overall thermal resistance as a function of a factor proportional to the water side thermal resistance. This factor contains $d^{0.2}$—the outside diameter to the 0.2 power. Since, for a given coil, the inner and outer diameter are constant, the water side thermal resistance is a function of only the water flow rate. Therefore, the R_0 intercept is independent of either d or d_i. It is assumed that in the calculation of the slope of the R_0 *vs* R_w lines the author determined the intercept from plots similar to that of Fig. 5. Thus, knowing R_a, R_w could be calculated for any water flow rate. A new plot of R_0 *vs* R_w could then be made and the slope of the lines measured. As a matter of interest this discusser would like to know why the water side resistance factor of Fig. 5 included $d^{0.2}$ rather than $d_i^{1.8}$, and furthermore, why the diameter was included in the factor at all.

The method of testing described in this paper was to establish heat balances within 5 percent and take q, the heat transfer rate, as q_w obtained from the water side "as this should be the highest value if there were any transfer of heat other than to the flowing air". It is realized that the possibility of heat transfer to materials other than the flowing air exists. However, the author reports that water temperature drops as low as 3.5 F were measured. Under this condition, an error of 0.1 F in the measurement of

the bulk water temperature could result in about a 3 percent error in the calculation of q_w. It has been this discusser's experience that in measuring small water temperatures extreme care must be taken to provide good mixing of the water stream at the point of temperature measurement. Furthermore, ordinarily small instrument errors can significantly effect accuracy. Did the author employ any special mixing device at the point of temperature measurement, and what type of instrumentation was used? Perhaps the use of either q_a or the average of q_a and q_w would have been more accurate than q_w where the water temperature drop was small.

In the determination of the overall mean temperature difference the author states that the coil was "visualized as a single tube in cross flow". Such being the case, this discusser wonders why the mean temperature difference was not taken as that for pure cross flow with one fluid completely mixed. The assumption of a single tube in cross flow would be justified if the fins were separated half way between the tubes. Since they were not, the temperature distribution in the fins was influenced by the adjacent tubes connected in series. Since each successive tube had a lower temperature than the one preceding it, a potential difference existed which tended to cause heat flow downward along the fins. This potential and heat flow depended upon the location of the fin in the core, and was not the same for each fin.

It is quite true that Equation 2 is recommended only to give values of water side heat transfer coefficients within 20 percent. Has the author considered the effects of the U-bends of the water side coefficient? Tests performed by this discusser have shown that the effects of U-bends on the water side coefficient are significant. For example, 3 coils were tested at the same air flow rate. These coils were of the same design except that the face dimensions were 12 in. x 12 in., 12 in. x 18 in., and 12 in. x 24 in. with the larger dimension in the direction parallel to the tubes. Tests on these coils (which had the same number of U-bends but different lengths of tubing) indicated that measurably higher overall heat transfer coefficients were obtained with the shortest coil at the same air flow rates. Thus the effect of the U-bends was to increase the effective water side heat transfer coefficient.

It was also found in tests conducted by this discusser that the plots of air pressure drop *vs* air weight flow resulted in curves which exhibited an upward trend at low air flow rates. It has been suggested that this may have been due to the entrance and exit losses becoming a larger percent of the total pressure drop at low air flow rates. It is encouraging to find that the author was able to correlate satisfactorily the isothermal pressure drop with that obtained on heating runs by using the density evaluated at the mean air film temperature which was taken as the arithmetic mean of absolute air and bulk water temperatures. By this method isothermal pressure drops can readily be converted to actual pressure drops for any operating condition.

The author is to be complimented for an interesting presentation of results. Future work is certainly to be encouraged, and this discusser shall be looking forward to reports on performance of 2-row coils and corregated one-row plate finned coils.

C. O. MACKEY, Ithaca, N. Y., (WRITTEN): The concept of a volumetric diameter has been used in the correlation of heat transfer data for the cross flow of fluids over tube banks* and compact exchangers. The volumetric diameter, d_v, may be defined as 4 times the ratio of the net air volume of the exchanger to the total air-side surface area of the fins and tubes. The number of fins per inch may be eliminated as a variable when volumetric diameter is used. Volumetric diameters of the coils described in this paper are given in Table A.

Empirical equations have been fitted to the experimental data for combined air-side unit conductances as follows:

$$\text{for } d_v G_0 < 6.3: \; h_a = \frac{2.4}{d_v} \left(\frac{d_v}{1}\right)^{0.58} (d_v G_0)^{0.400 - 0.043} S_t$$

* ASME Transactions, by O. P. Bergelin, G. A. Brown, H. L. Hull and F. W. Sullivan (Vol. 72, 1950)

$$\text{for } d_v G_0 > 6.3: h_a = \frac{1.8}{d_v}\left(\frac{d_v}{l}\right)^{0.58}(d_v G_0)^{0.556-0.043 S_t}$$

$$\text{where } l = \text{fin depth, in.; and } S_t = \frac{p}{d}$$

These equations will fit most of the experimental data with an error that is commonly less than 15 percent.

The contact-mixture analogy† has been used by many engineers interested in coil performance when there is simultaneous cooling and dehumidifying. In terms of the air-side surface conductance, the contact factor of a one-row coil is

$$C_1 = 1 - e^{(-h_a A_a)/60 G_0 A_0 C_p}$$

TABLE A—VOLUMETRIC DIAMETERS OF TEST COILS

COIL NO.	d_v, IN.	COIL NO.	d_v, IN.	COIL NO.	d_v, IN.
1	0.530	13	0.600	32	0.246
2	0.330	14	0.354	33	0.155
3	0.242	15	0.254	34	0.156
4	0.190	16	0.200	35	0.259
5	0.159	17	0.161	36	0.600
6	——	20	0.578	37	0.616
7	0.584	21	0.250	38	0.262
8	0.256	22	0.156	39	0.162
9	0.159	28	0.546	40	0.540
10	0.565	29	0.244	45	0.529
11	0.257	30	0.152	46	0.530
12	0.162	31'	0.540		

As an example, assume that it is desired to find the contact factor for coil number 38 for $G = 50$ lb per (min) (sq ft).

From the empirical equation just given, the air-side surface conductance is

$$h_a = \frac{1.8}{0.262}\left(\frac{0.262}{2}\right)^{0.58}(13.1)^{0.556-0.043(5.18)}$$

$$= 5.0 \text{ Btu per (hr) (sq ft) (F)}$$

(Note: Fig. 7 gives a result that is about 8 percent lower than that given by the empirical equation).

† ASME Transactions, by W. H. Carrier (Vol. 59, 1937).

For $C_p = 0.243$ Btu per (lb)(deg F) lb F (*moist* air), the one-row contact factor of this coil at this flow rate is

$$C_1 = 1 - e^{[-5(37.77)]/60(50)(1.361)(0.243)} = 0.173$$

A measure of overall coil performance, not specifically included in this paper, is a plot of the air-side conductance against the horsepower required to overcome the pressure drop *per square foot of air-side surface.* In Fig. 12, the coils with the most air-side surface appear to have a poor performance. Actually, the use of sufficient surface with a coil like 36 to get a required rate of heat transfer with an available mean temperature difference may result in about the same friction horsepower for some values of G_0 as when a coil like 17 is used for the same capacity. This is illustrated in Table B which compares coils 36 and 17.

TABLE B—COMPARISON OF COILS 36 AND 17

COIL NO.	G_0	k_a	HORSEPOWER PER SQUARE FOOT OF AIR-SIDE SURFACE
36	30	5.9	$6.3(10)^{-5}$
	50	7.2	$2.9(10)^{-4}$
	70	8.1	$7.2(10)^{-4}$
	100	9.3	$2.0(10)^{-3}$
17	30	6.0	$7.6(10)^{-5}$
	50	7.0	$2.9(10)^{-4}$
	70	7.4	$6.9(10)^{-4}$
	100	8.1	$1.6(10)^{-3}$

H. B. NOTTAGE, Encino, Calif.: To a partial extent, my comments will parallel those of the preceding discussion, but the points are important, nevertheless.

The mean temperature difference is a tantalizing proposition. While the author has obtained a workable temperature difference for interpolation, this certainly does not answer any fundamental questions as to what may be taking place inside the heat exchanger. In tests like this, expediency leads to treating the heat exchanger as a sort of *black box*, with fluid streams (or energy streams) entering and leaving with measured terminal states. Without really understanding what happens inside the *black box*, crude overall factors are calculated.

Yet, within the *black box*, conditions vary significantly from point to point. Fluid temperatures, metal temperatures, heat transfer rates, and heat transfer resistances all vary in some microscopically-unknown manner. Using terminal fluid temperatures and a geometrical surface area one then computes an overall conductance, or resistance, which is the net integrated effect of each and every point-to-point condition throughout the exchanger. Is it then any wonder that such crudities can be *correlated* (how defined?) only by empirical *oddities* such as θ_m?

To obtain further evidence on the question of a *mean temperature difference* (how should one define a mean?), it is recommended that the same coils be tested under cooling conditions.

These comments, please, are not intended to detract from the practical utility of any sort of *correlation* which serves to interpolate performance in simple (?) *black box* terms; for such data enable us to obtain useful numerical answers.

Next, in the various possible comparisons of coils in terms of pressure drop, a pumping power, and heat transfer rate, I feel that a missing consideration is some sort of useful measure of the *dollar cost* of purchasing and operating a coil. Among possible considerations, what might you recommend?

Then, would the author explain in further detail the performance differences observed when the asymmetrical fins were reversed? What quantitative examination has been made of fin effectiveness variations, flow eddies behind the tubes, and entrance flow over the fins treated as flat plates?

Finally, what has been done towards establishing and analyzing the constituent effects in the overall pressure drop? I have reference here to flow contractions, direction turnings, eddies, flow expansions, and the effects of changes in fluid density.

Continuing study is awaited with sincere interest. Will you seek to get inside of the *black box* experimentally?

TABLE C

	q = Btu/hr-ft² coil face-°F MTD	
	$q_0(h_w = \infty ; G_0 = 40)$	$q(h_w = 50; G_0 = 40)$
Coil 32	138	39
Coil 34	163	23

DAN KRAMER, Trenton, N. J.: Professor Shepherd's paper on finned coil performance is the most comprehensive experimental study on this subject it has been our privilege to examine. Prof. Shepherd must be complimented on his patience, tenacity and fine workmanship.

Prof. Shepherd's paper as printed, omits the information that q_0 is in every case the heat transfer which would be expected were the internal coefficient infinite.

For this reason the curves which plot values or functions of q_0 and the conclusions drawn therefrom, though undoubtedly sound, must be employed with caution since the application of internal coefficients within the normal working range will result in repositioning and even reversal of the relative positions of some of the curves as indicated by Table C.

Since the intersection of the curves for coils 32 and 34 on h_w *vs* q coordinates takes place at a value of h_w greater than 1000, coil 32 will have a greater capacity than coil 34 throughout the useful range of internal coefficients, exactly opposite to the relationship indicated in Fig. 11.

This discussion is emphatically not intended as rebuttal to Prof. Shepherd's paper but merely to emphasize the caution with which the obviously sound and clearly presented data must be used.

AUTHOR'S CLOSURE: I wish to thank all those who contributed to the discussion, particularly Mr. Sartain for his detailed comments and queries, and Professor Mackey for his correlation and use of the data in terms of volumetric diameter. I am also indebted

to a correspondent who pointed out certain discrepancies in Figs. 7 and 11 as they appeared in the JOURNAL SECTION and the reprints. An error in replotting for publication had occurred in (A) of Fig. 7 and this has now been rectified in the figure published herewith. Correct values had been used for the further calculations of q_0, so the remainder of the figures are valid. In Fig. 11, the values of q_0 for coil 6 should be divided by 1.361 (coil face area), as these are in error as printed herewith.

Concerning Mr. Sartain's discussion I have the following comments.

(1) It is correct that the R_0 intercept is independent of the tube diameter but $d_i^{1.8}$ was included in Equation 3 in order that the values of R_w should be on a comparable basis for all tube sizes. The $d^{0.2}$ factor in Fig. 5 should be replaced by $d_i^{1.8}$ and this again results from an error in replotting for the published figures.

(2) The measurement of water temperature was given great consideration. Insulated mixing chambers effecting high turbulence were provided at inlet and outlet, with the temperatures being read by mercury thermometers immersed in oil wells. Originally, the temperature drop was measured by a five-element differential thermopile, but finally it was found more convenient for a single operator to take the data in the manner indicated. The differential thermopile method was useful in checking directly the accuracy of the thermometer method, as the stems of the thermometers were used as cores on which to secure the thermocouple elements. For the extreme low drop of 3.5 F, there is a possible error of 0.2 F, but here the heat balance is a check.

(3) The effect of U-bends on the internal coefficient is undoubtedly considerable, but the values of h_w obtained by the author were somewhat lower than those given by Equation 2, not greatly so, but sufficient to rule out the use of the equation in its entirety.

(4) Any use of an overall mean temperature difference is in my opinion only thoroughly justified for a true longitudinal counter-flow exchanger. In the case of cross-flow, it ceases to have an exact meaning, because of the assumptions of *mixed* and *unmixed* flow which are required in the customary calculated correction factors applied to the counter-flow MTD. It would seem to have true validity for use in calculating heat exchanger performance only when there is geometric and hydrodynamic similarity between the exchanger in question and the prototype from which the design data, which are being used, were obtained. Nevertheless, it is the only useful correlating parameter which is available and providing it is used in a corresponding manner, its empirical nature is justified. To this extent, it might be likened to the *characteristic dimension* used in the Reynolds number, which may be arbitrary, and which should not necessarily be used as giving an absolute value.

(5) For air horsepower, etc., this is dealt with in the reply to Professor Mackey's discussion.

The following apply to Dr. Nottage's discussion. With respect to the *black box*, I am sure we would all like to get inside it and discover exactly what was taking place. The paper as given is an effort only to present the bare data in as useful a form as possible for engineering purposes. However, a considerable amount of analysis has been done, which it was not possible to present here. The apparatus was not suitably instrumented to obtain all the desirable information, but from tests on corresponding coils with no fins, it was possible to attempt a breakdown of the heat transfer and pressure drop as occurring over flat plates and over tubes in cross-flow. To do this, the fin effectiveness was calculated by means of relaxation methods. A fair amount of success was achieved with the pressure loss data, so that a value for any tube spacing, fin spacing and fin size could be calculated, the resulting expression giving values within about 12 percent for any coil within the range tested and within 5 percent for the majority. The heat transfer data were less tractable and the experimental data for the bare tubes, other than coil 6 as reported, were not sufficiently accurate to warrant much consideration as the temperature changes were so small. Since this work was done, considerable attention has been paid in this laboratory to the question of the *black box* in various manifestations and these are continuing. Discussions of mean temperature difference and dollar cost are given in reply to the other discussion.

In reply to Professor Mackey's remarks, I offer the following comments.

The correlation in terms of volumetric diameter is very helpful and the example of its use in conjunction with the contact-mixture analogy provides a useful illustration. Like mean temperature difference, the volumetric diameter cannot be said to have an absolute significance, but it is often a very useful tool in presenting data for practical engineering use. The presentation of conductance and horsepower per square foot of air-side surface is another good method of measuring overall coil performance, a matter which concerns other discussers. It brings in running cost, as power absorbed, and capital cost to the extent of surface required. The *dollar cost*, while certainly the basic consideration for heat exchanger work of this nature, is very elusive and there does not yet appear to be a criterion of any really wide utility. It is hoped that the data in the simple form presented here, will enable those interested to form their own performance factors to suit their particular need.

Concerning the remarks of Mr. Kramer, the paper is thought to be clear in indicating that the values given for q_0 are for the air side alone, that is, with an infinite internal coefficient, since this was the method of determination of the h_a values on which q_0 is based. However, Mr. Kramer's comments and examples are useful in emphasizing the fact that an overall rate heat transfer in a given exchanger is a function both of the coefficients and of the surface areas on each side. With coils of the type reported, having a large value of the ratio of external to internal surface area (ranging from about 4 to 1 to 75 to 1), the internal coefficient must certainly be considered if it is not infinite.

1587

No. 1587

TEMPERATURE AND HUMIDITY EFFECT ON ODOR PERCEPTION

By W. F. KERKA* AND C. M. HUMPHREYS**, CLEVELAND, OHIO

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS at its Research Laboratory located at 7218 Euclid Ave., Cleveland 3, Ohio

THIS paper is the first of a series dealing with odor problems in air-conditioned spaces. The purpose of this study was to determine the effect of temperature and humidity upon odor perception. Odor-level evaluations of 3 pure vapors and of cigarette smoke were made using the organoleptic-panel technique. The results were plotted on the psychrometric chart and perception trends were determined. The effect of olfactory adaptation was studied by exposing panel members to a fixed concentration of odorant, and determining the perception levels at specified intervals of time.

The perception trends thus established serve only as a guide, and restrictions should be placed upon them when considering their applicability to an actual case. The concentration of odorants outdoors as well as in occupied spaces varies with changes in temperature and humidity. As an odor source is heated, it gives off more odorant, and in air-conditioned spaces where dehumidification takes place, water soluble odorants are removed along with the condensate on the surfaces of the coil. Hence what may be mistaken for effects of temperature and humidity upon odor perception may actually be variations in odor concentration. The study of the effect of temperature and humidity upon odor perception was conducted with constant concentrations of odorants.

TEST APPARATUS

Test Rooms: Odor appraisals were made in 2 test rooms[1], separated by a corridor, and located in the main Laboratory of the ASHAE Research Building. The rooms

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[1] Exponent numerals refer to Bibliography.

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measured 8 x 12 x 8 ft high and were lined with porcelain enameled steel having low odor-adsorption qualities. Test-room doors were weatherstripped to reduce infiltration of outside air. Inlet air was filtered and deodorized by passing through 2 banks of mechanical filters, an electrostatic precipitator, 2 banks of activated carbon canisters, and finally through an absolute filter. Air from the rooms was not recirculated, but was exhausted to the outdoors. Three separate systems, serving the 2 rooms and the corridor, contained the necessary equipment for cooling, heating, and humidification. Air flow was regulated by adjustable dampers on the inlet side of the supply fans, and metering orifices were used to measure the flow rates. The odorant was introduced into the air-supply duct and thoroughly mixed with the air stream by 4 baffles.

Odor Introduction Apparatus: The apparatus used for the introduction of pure odorous vapors consisted of 2 gas-washing bottles in series through which nitrogen

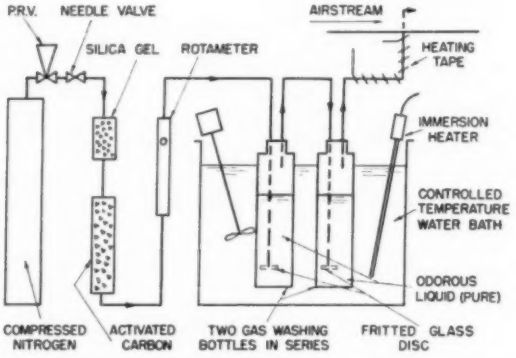

FIG. 1—SCHEMATIC DRAWING OF ODOR INTRODUCTION APPARATUS (PURE VAPORS)

was bubbled (Fig. 1). Nitrogen was used in place of air since it was less likely to react chemically with the odorant. The nitrogen was passed through beds of silica gel and activated carbon to remove inherent moisture and odor before it was metered by a rotameter into the bubblers.

Static pressure at the inlet to the rotameter measured 0.5 in. Hg for a flow rate of 15 ml per min and 0.6 in. for a flow rate of 1.2 liters per min, the range used in the perception studies. The nitrogen leaving the second bottle was saturated with vapor and entered the air-supply duct via a tube, heated with a resistance tape to prevent condensation. The apparatus was constructed of glass, and ground joints were used (after the bed of activated carbon) to prevent contamination. The gas-washing bottles were surrounded by a controlled temperature water bath. The concentration of odorant was calculated from its vapor-pressure properties, the amount of nitrogen bubbled through the washers, and the quantity of air supplied to the test room. The concentration was checked by adsorbing the odorant on activated carbon and measuring the increase in weight.

The apparatus for producing cigarette smoke used the same components as the one just described, except that a bell jar was substituted for the washing bottles, and compressed air was used in place of the nitrogen. The cigarettes were allowed to burn freely on a screen platform suspended 3 in. above the base of the jar which rested in a pan of water acting as an air seal. Combustion air was admitted through a tube at the base of the jar, passed around the cigarettes, and the combustion products were removed through a 1-in. steel tube leading to the air-supply duct. The tube was heated to prevent condensation of the products of combustion. In a second smoke-producing method, the cigarettes were allowed to burn freely in the test rooms on a screen platform. In a third method, human smokers, seated in the rooms, puffed at their normal rate. A maximum of 3 smokers was employed. The same popular brand of king-size cigarettes was used for all test runs.

Temperatures and humidities were determined from the readings of wet- and dry-bulb thermocouples located in each room. The wet-bulb thermocouple was made by inserting a copper-constantan junction into a ¼-in. brass tube filled with solder. A wetted wick surrounded the tube and extended along the leads for about 1½ in.

TABLE 1—SENSORY SCALE FOR EVALUATING INTENSITY LEVELS

DEGREE OF ODOR INTENSITY	DESCRIPTION
0	Odorless
1	Threshold
2	Definite
3	Strong
4	Overpowering

The Organoleptic Panel: The organoleptic panel technique[2] was used in making judgments of odor intensity and quality. The panel consisted of 6 members from a staff of 12 people who had been trained in odor appraisal for several months before the beginning of the test runs. The panel was about equally divided between men and women, and moderate smokers and non-smokers. Four of the panel members served continuously, while the other 2 were selected from the remaining staff on the basis of their availability at the time of the test. Subjects were requested not to smoke less than ½ hr before each test. Persons having colds were excused from the panel and alternates used in their places.

TEST PROCEDURE

In evaluating the intensity levels for pure vapors, the sensory scale of Table 1 used by Gee[3] was adopted. The score of 1 indicated a recognition *threshold*. A score of ½ was used when the panel member detected the presence of an odor, but felt its intensity was not great enough to identify its quality. An intensity level between *threshold* and *definite* was scored 1½, and so on.

When preliminary tests on tobacco smoke were begun, the panel members felt that the word description of the sensory scale used for pure vapors did not sufficiently

represent their reactions. As a result the sensory scale used by Yaglou[4] (Table 2) was adopted for all cigarette-smoke evaluations.

Subjects recorded separately the degree of odor and irritation, also place of irritation (eyes, nose, or throat).

Selection of Odorants: The 3 pure vapors selected for investigation were iso-valeric acid, methyl salicylate, and pyridine. Their odors respectively are sour and footy, fragrant and cool, and burnt and nauseating. During all of the tests, the concentration of each odorant was held constant and was the same in each room.

The cigarette smoke was generated in the bell jars and introduced into the air-supply ducts at a constant rate, controlled by the number of burning cigarettes and the amount of combustion air. During all of the test runs (pure vapors and cigarette smoke) the ventilating rate to each room was fixed and only the temperatures and humidities were varied.

TABLE 2—SENSORY SCALE FOR TOBACCO-SMOKE EVALUATIONS

DEGREE, ODOR OR IRRITATION	DESCRIPTION
0	*Imperceptible* odor, or irritation.
1	Perceptible odor, or irritation, but *not objectionable.*
2	Moderate odor, or irritation, little or no objection, *acceptable* level.
3	*Objectionable* odor, or irritation; condition regarded with disfavor.
4	Strong odor, or irritation, but *endurable.*
5	Very strong odor, or irritation, *intolerable.*

Panel Technique: When the desired test conditions were established in the rooms, subjects were called from their normal work routines (calculating, typing) to make odor appraisals. Each subject entered a test room alone, normally clothed, and without knowing whether the conditions would be more or less humid than in the previous test. After walking to the center of the room, the person sniffed for a first impression, then exhaled, and about 5 sec later sniffed again for a second impression. He then circled a number indicating his judgment of the odor intensity and also noted his impression of the odor quality on a form which he carried. In tests with tobacco smoke, the degree of irritation was also noted. After a rest of about 2 min in the Laboratory, the subject entered the second test room and repeated the foregoing procedure. The slip bearing the appraisals for the 2 tests was submitted to the test engineer. Subjects were asked not to discuss their judgments with each other, to prevent the impressions of one being influenced by the impressions of another.

It was possible for the judgments in the second room entered to be influenced by adaptation to the odor. To eliminate this effect, during each test half of the panel members first entered Room No. 1 and then Room No. 2, while the other half entered the rooms in the reverse order. Also the conditions in the rooms were changed after each run so that the odor level at a particular temperature and humidity judged in one room was also judged in the other. At least 1 hr was allowed between each run and no more than 6 runs were made each day.

The conditions selected for study ranged from 55 F to 110 F (dry bulb) and 20 to 80 percent RH (relative humidity) (10 to 40 percent above 95 F). The test pattern was set up so that one dry-bulb temperature was selected each day and the relative humidities varied by 10 percent intervals. Table 3 illustrates a test pattern.

The same constant dry-bulb temperature and odor concentration was maintained in both rooms during all 6 runs. A run consisted of 6 appraisals made in each of the 2 rooms.

During some test series, the relative humidities were selected in decreasing rather than increasing order. The dry-bulb temperatures themselves were chosen in random order. The panel member was never told what test condition prevailed.

A possible shift in the appraisals of the panel members over a period of time could give misleading results, especially if an orderly sequence of conditions were investigated. For this reason during each of the temperature-humidity *vs* percep-

TABLE 3—EXAMPLE OF TEST PATTERN

RUN NO.	RELATIVE HUMIDITY	
	ROOM 1	ROOM 2
1	30	40
2	40	50
3	50	60
4	60	70
5	70	80
6	80	30

tion level studies, the dry-bulb temperatures were investigated at random rather than in consecutive order. At regular intervals, extra runs were made (unknown to the panel members) at concentrations other than the fixed value used during the tests. This was done to check the responsiveness of the panel members and to reorient them in their appraisals.

It was originally planned to use the central corridor as a preconditioning chamber; but, at the beginning of the program the panel members reported that this area became contaminated with odorant as a result of opening the doors to the test rooms, even though a high ventilating rate was used in the corridor. Since any exposure to the odorant, immediately preceding an evaluation, could not be permitted, the method was abandoned and the subjects entered the test rooms directly from the Laboratory. All of the tests were conducted during the winter and early spring. Daily measurements in the Laboratory showed a fairly constant condition of 76 F dry-bulb and 30 percent relative humidity. The panel members remained in this environment for at least a minute before making their first appraisal. Since the static pressure in the test rooms was maintained at a few hundredths of an inch of water less than in the Laboratory, contamination of the surroundings was no serious problem under normal operation. After each day of testing, the rooms were cleansed by admitting humid air at a dew-point temperature above the temperature of the wall surfaces, and then evaporating the resulting film deposited on the surfaces by introducing warm dry air. The floors were washed

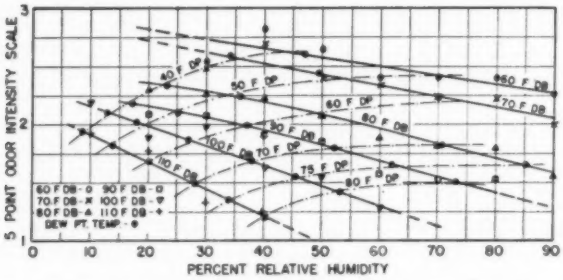
5 POINT ODOR INTENSITY SCALE
PERCENT RELATIVE HUMIDITY
40 F DP
50 F DP
60 F DP
70 F DP
80 F DP
90 F DP
100 F DP
110 F DP
50 F DB
70 F DB
80 F DB
90 F DB
75 F DP
60 F DB - O 90 F DB - □
70 F DB - X 100 F DB - ▽
80 F DB - Δ 110 F DB - ◇
DEW PT. TEMP - ⊕

FIG. 2—ODOR INTENSITY VS PERCENT RELATIVE HUMIDITY (CONSTANT TEMPERATURE LINES FOR CIGARETTE SMOKE)

daily with warm water and detergent and periodically the walls were wiped by the same method.

Equilibrium Conditions: During the test runs to establish perception trends for pure vapors, the air flow and admission of odor were maintained at constant values throughout a day of testing. Since the air and odorant distributed to each room equaled the amount exhausted, a continuous steady-state was possible. During the cigarette-smoke tests, the free-burning time for king-size cigarettes in the bell jar was about 25½ min. The cigarettes were lighted and allowed to burn for 10 min, and then appraisals were made during the last 15 min of the burning period. At the end of each run, the jars and screen platforms were cleaned of residue ash. In the tests where the cigarettes were humanly smoked or allowed to burn freely in the test rooms, a smoking period of about one-half hour was allowed before evaluations were made.

Treatment of Data: Tabulations of the arithmetic mean scores (each score being the mean of 12 appraisals) for the odor intensities of iso-valeric acid, methyl salicy-

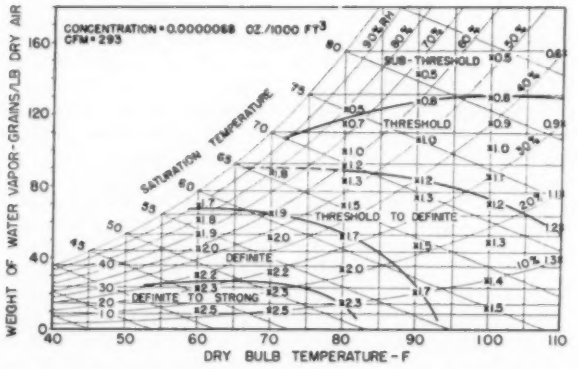

FIG. 3—ODOR PERCEPTION TRENDS FOR ISO-VALERIC ACID

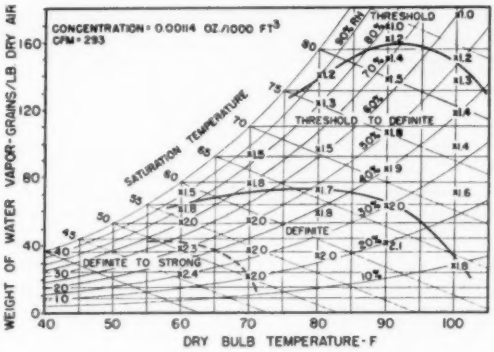

FIG. 4—ODOR PERCEPTION TRENDS FOR METHYL SALICYLATE

late, pyridine, and cigarette smoke were made, and were based on the initial impression of odor level. These tabulated mean scores were used in plotting constant dry-bulb temperature lines, illustrated for tobacco smoke in Fig. 2. As an example, the values for 70 F dry bulb in Fig. 2 were obtained by averaging the mean scores at 65 F, 70 F, and 75 F. The odor level scores, obtained from the intersection of the constant-temperature and the relative-humidity lines, were plotted on the psychrometric chart, Figs. 3 through 6. From these values, the perception-intensity areas were charted. This method of cross plotting was used since the individual scores were not sufficiently consistent for a direct interpretation.

A tabulation of the median scores was also made. These were plotted on graphs identical to Fig. 2 and constant dry-bulb temperature lines were drawn. The

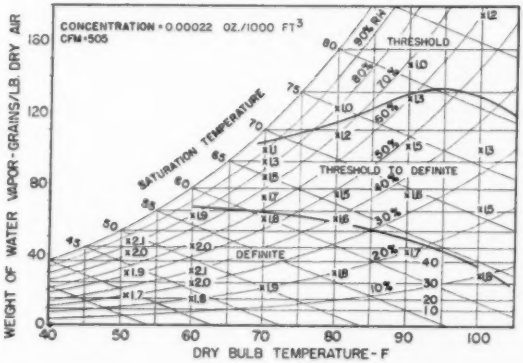

FIG. 5—ODOR PERCEPTION TRENDS FOR PYRIDINE

trend and displacement of these lines were substantially the same as the ones obtained from the arithmetical mean scores; however, the median scores were less consistent than the means and it was more difficult to fit a curve to a series of points.

DISCUSSION OF RESULTS

The perception areas for iso-valeric acid were drawn as three bands in Fig. 3. The results indicate that the perception level for iso-valeric acid decreases as the relative humidity increases at a constant dry-bulb temperature. This trend becomes more evident at the higher temperatures investigated. The effect of temperature alone can be determined by moving along constant dew-point temperature (constant specific humidity) lines. Below about 70 F dew-point the odor level falls off with an increase in dry-bulb temperature. At higher dew-points, the perception level appears to rise slightly with an increase in temperature.

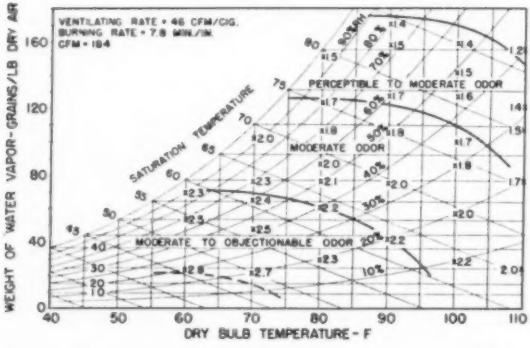
VENTILATING RATE = 46 CFM/CIG.
BURNING RATE = 7.8 MIN./IN.
CFM = 184
SATURATION TEMPERATURE
PERCEPTIBLE TO MODERATE ODOR
MODERATE ODOR
MODERATE TO OBJECTIONABLE ODOR
WEIGHT OF WATER VAPOR-GRAINS/LB DRY AIR
DRY BULB TEMPERATURE - F

FIG. 6—ODOR PERCEPTION TRENDS FOR CIGARETTE SMOKE

The perception trends for methyl salicylate are shown in Fig. 4. It was found that some of the individual scores between 55 F and 70 F dry-bulb, first rose, then fell, with an increase in relative humidity from 30 percent to 80 percent. Statistical analysis has been applied to these data since the paper was first published in the JOURNAL SECTION of *Heating, Piping & Air Conditioning*. It was found that this apparent reversal was not significant when treated as both paired and unpaired data for small samples. Above 75 F dry bulb, the perception-level scores decreased rather uniformly with an increase in relative humidity at constant dry-bulb temperature. The data were too inconsistent to show any definite effect of temperature upon odor perception. If the values at 90 F dry-bulb (Fig. 4) were more in line with the values at 80 F and 100 F, however, it would appear that an increase in temperature at constant specific humidity would cause a general decrease in perception level. The lines separating the perception areas are shown broken where their position is open to question.

The perception areas for pyridine are shown in Fig. 5. At low dry-bulb temperatures (47 F to 65 F), the scores indicated an increase in perception level with an increase in relative humidity to about 40 percent. Again, since the paper was

first published, statistical analysis using paired and unpaired data was applied, and this reversal of the odor-level trend below 65F was found to be insignificant. Above 75 F dry bulb, however, the odor intensities decreased rather uniformly with an increase in relative humidity at constant dry-bulb temperature. An increase in temperature at constant specific humidity causes a slight drop in perception below about 60 F dew-point. Above this value the effect appears to reverse itself.

The odor perception areas for cigarette smoke, Fig. 6, form 3 distinct bands. Figs. 2 and 6 indicate a general decrease in odor perception level with an increase in relative humidity at a constant dry-bulb temperature. The trend becomes more evident at the higher temperatures investigated. An increase in dry-bulb temperature at constant specific humidity also causes a decrease in perception level. This drop in level becomes more noticeable as the temperature increases. The values in Fig. 6 indicate odor intensity only. Initial effects of irritation either to the

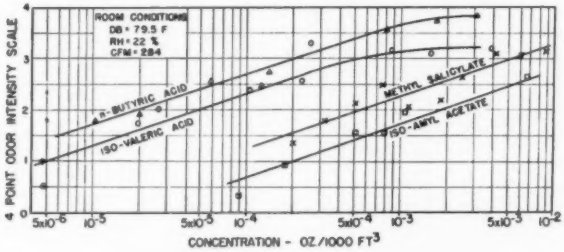

FIG. 7—ODOR INTENSITY VS CONCENTRATION

eyes, nose, or throat were recorded separately, but the results were less consistent than the odor intensity scores, although the same general trend was followed.

In Figs. 3 through 5 the lines separating the perception areas do not represent sharp cleavages in odor levels, but serve only as general boundaries between transition scores. Examination of the perception trends for the 3 pure vapors reveals that the odor intensity of iso-valeric acid drops more rapidly with an increase in relative humidity at constant dry-bulb temperature than either methyl salicylate or pyridine.

Physical Significance of Perception Trends: The words used in the sensory scales, as indications of odor intensity, can be given a very real and significant meaning if odor level evaluations are made over a wide range of odorant concentration. Fig. 7 shows evaluations of iso-valeric acid and methyl salicylate and 2 other pure vapors, n-butyric acid and iso-amyl acetate, plotted against the logarithm of the concentration. During these tests, the conditions were maintained at 79.5 F dry-bulb and 22 percent RH. The air supply was held at 284 cfm (22 air changes per hour). Pyridine was not evaluated as its odor was too nauseating at higher concentrations. Fig. 7 indicates that the physical concentration must be increased tenfold to raise the odor intensity by one point. Hence, to raise the score 2 points in odor level, requires a hundredfold increase in concentration. This is in accordance with the

Weber-Fechner Law. The relationship appears to hold true for the mid-point of the odor scale but may deviate at the ends[5].

Therefore, in Figs. 3 through 5, any increase in humidity at a given dry-bulb temperature that lowers the odor intensity from a *definite* to a *threshold* level, has the same effect as reducing the physical concentration of odorant by 90 percent. Likewise, any decrease in odor level of one-half point due to either an increase in humidity at constant dry-bulb temperature or án increase in temperature at constant specific humidity has the same effect as decreasing the physical concentration of odorant by about 67 percent.

The physical significance of the perception areas for cigarette smoke is shown by Line A in Fig. 8. The ventilating rate must be increased about sixfold to reduce the

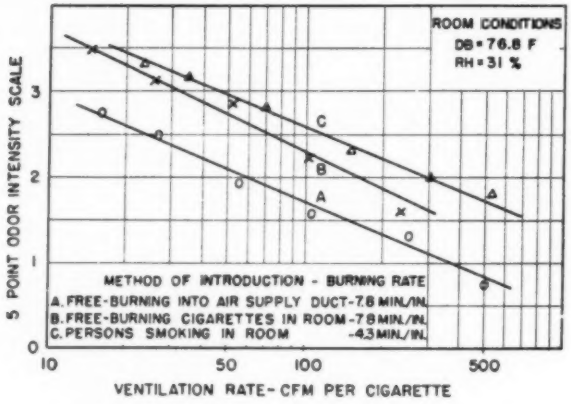

FIG. 8—ODOR INTENSITY OF CIGARETTE SMOKE VS VENTILATION RATE

odor level by a full point. The area of *perceptible* to *moderate* odor level in Fig. 6, therefore, has an apparent ventilating rate 6 times as great as the *moderate* to *objectionable* odor level.

It should be pointed out that, if the sensory scale used for pure vapors were superimposed on Fig. 8, the ventilating rate for Line A would then have to be increased tenfold for a one-point drop in odor level of the four-point scale. This is compatible with the results for pure vapors, since 1.3 points on the tobacco-smoke sensory scale equals one point on the pure-vapor scale.

A discrepancy occurs between the odor level evaluations of iso-valeric acid in Figs. 3 and 7. At a concentration of 0.0000068 oz. per 1000 cu ft, the intensity was judged at about *definite* (80 F and 20 percent RH) in Fig. 3, while in Fig. 7 at the same concentration it was rated at about *threshold*. It is possible for a shift in judgments to occur. In all probability the points for iso-valeric acid in Fig. 7 could be moved to the left, at the same slope, however.

Effect of Temperature and Humidity Upon Odor Quality: During the perception-trend studies, the panel members reported that little if any change in odor quality

occurred over the range of temperatures and humidities investigated. The sensation of the inhalation appeared to be *heavy* at low temperatures, however, and *lighter* at high temperatures.

Odor Adaptation: During the perception-trend studies the impression of the first inhalation experienced by a subject was in the majority of cases higher than the impression of the second inhalation. This would indicate that adaptation to odors takes place with the first inhalation. Several panel members reported from time to time that the strength of an odor diminished during an extended initial sniff.

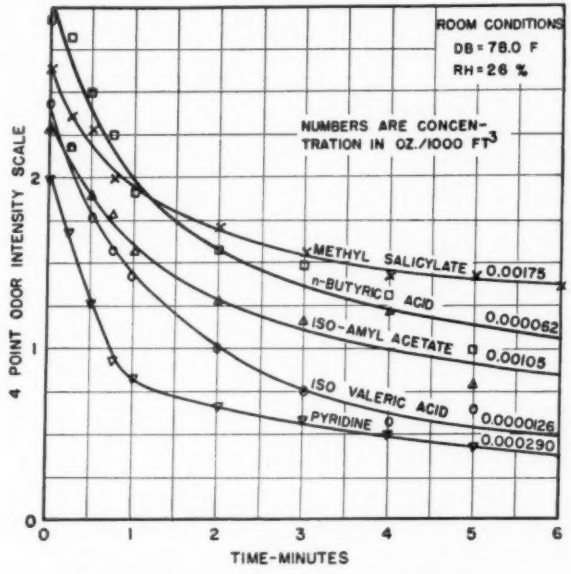

FIG. 9—ADAPTATION TO PURE-VAPOR ODORS

To illustrate the effect of odor adaptation, the panel members were exposed to a constant physical concentration of odorant for a given period of time. The member entered the test room and sniffed immediately for an initial impression. Breathing normally through the nose, judgments were made every 15 sec for the first minute and every minute thereafter for a maximum of 6 min.

Fig. 9 gives the results for each of the 3 pure vapors plus iso-amyl acetate, and n-butyric acid. Each point represents the mean score of 6 to 8 judges. The concentration was adjusted to give initial readings between 2 (*definite*) and 3 (*strong*). In all of the tests, adaptation was greatest during the first stages of exposure. The curves for iso-valeric acid and n-butyric acid are nearly the same shape (although displaced), indicating an identical adaptation rate. In all of the tests the conditions were maintained at 78 F dry-bulb and 26 percent RH.

Adaptation to Cigarette Smoke: Figs. 10 and 11 give the evaluations for exposure to cigarette smoke under 30, 47, and 65 percent RH and at the same dry-bulb temperature of 77 F. For each humidity the smoke was introduced into the rooms by each of 2 methods: (*a*) free-burning directly in the room (Fig. 10), and (*b*) free-burning in the bell jars with the smoke introduced into the air-duct system (Fig. 11). The ventilation rate was maintained at 14 cfm per cigarette for both methods, and the burning rates were the same (25½ min per cigarette). Besides recording odor levels at given intervals of time, the panel members evaluated effects of irritation either to the eyes, nose, or throat.

The results show a higher rating of irritation and odor level at all 3 humidities when the panel members were exposed to the smoke generated directly in the rooms.

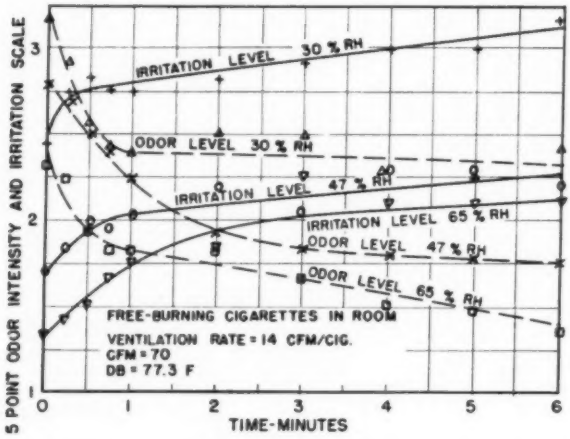
5 POINT ODOR INTENSITY AND IRRITATION SCALE
TIME-MINUTES
IRRITATION LEVEL 30 % RH
ODOR LEVEL 30 % RH
IRRITATION LEVEL 47 % RH
IRRITATION LEVEL 65 % RH
ODOR LEVEL 47 % RH
ODOR LEVEL 65 % RH
FREE-BURNING CIGARETTES IN ROOM
VENTILATION RATE = 14 CFM/CIG.
CFM = 70
DB = 77.3 F

FIG. 10—ADAPTATION TO CIGARETTE-SMOKE ODOR, AND IRRITATION CAUSED BY CIGARETTE SMOKE GENERATED IN ROOM

This points to the removal of some of the odorous and irritating components by condensation or adsorption on the surfaces of the air duct, when the smoke was introduced into the air stream.

Under the 30 percent RH condition, odor levels and irritation levels (confined mostly to the eyes and nose) were greater than the odor and irritation levels at 47 and 65 percent RH. This was noted for both methods of cigarette-smoke introduction.

General Discussion of Results—Cigarette Smoke: The problem of mechanically producing measurable quantities of tobacco smoke, which is representative of human puffing, has no ready solution. The nicotine content in the smoke varies with the strength of puff, moisture content in the tobacco, and length of the unburned portion[6]. Yaglou[4] describes the effect of nicotine on both smokers and nonsmokers; his use of actual persons smoking at normal rates is perhaps the most realistic approach to the problem.

While the authors admit the free-burning method does not duplicate human action, it does afford a simple and economical means of generating reproducible quantities of smoke. Kuehner[7] states that artificial burning, regardless of combustion rate, degree of freshness of tobacco, or brand, results in a uniform production of odor per unit weight of tobacco burned, and that a constant difference exists between human and mechanical smoking because of the absorption of odor in the respiratory system.

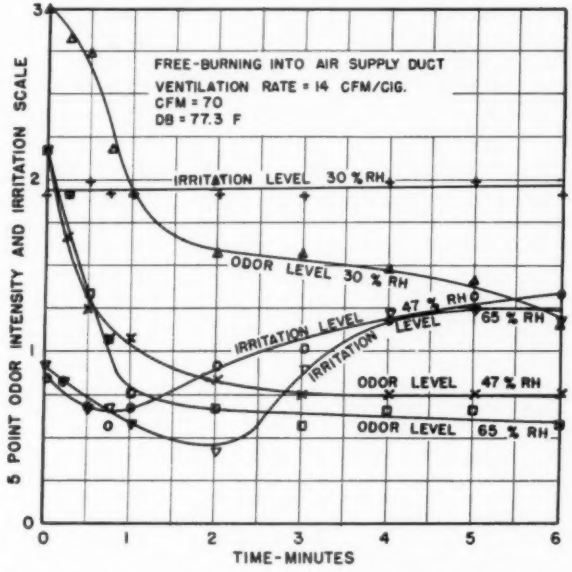

FIG. 11—ADAPTATION TO CIGARETTE SMOKE ODOR, AND IRRITATION CAUSED BY CIGARETTE SMOKE INTRODUCED WITH SUPPLY AIR

The perception trends for cigarette smoke (Fig. 6), although not derived under conditions that duplicate human smoking, could be adjusted in value to represent such a condition. In Fig. 8 the difference in odor level between cigarette smoke introduced into the air-supply duct (Line A) and smoke produced by human smokers (Line C) is evident. At all values of cfm per cigarette the odor level of Line C is equally displaced above Line A. Line B represents evaluations of the odor level of tobacco smoke generated by free-burning cigarettes located in the rooms. The displacement of Line B above Line A again points to the adsorption and condensation of odorous components on the surfaces of the air duct. The burning time for cigarettes used in the evaluations of Lines A and B was 25.5 min per cigarette or

7.8 min per in. The burning rate for the cigarettes smoked by persons (Line C) was about 4.3 min per in. Although some of the odor is absorbed in the respiratory system, the greater amount of tobacco burned per minute, and possibly the inhalation of air through the unburned portion, produces a higher odor level than the free-burning method at the same ventilating rate. The displacement of Line C above Line B appears to confirm this. In order to compare the evaluations of Line C with Line B on the basis of cubic feet per minute per weight of cigarette burned in a given period of time, Line C would be displaced to the left by the ratio of the burning rates.

From the results shown in Fig. 8, the perception areas for cigarette smoke in Fig. 6 could each be rated about one point higher had the evaluations been made on tobacco odors produced by persons smoking in the test rooms at the same ventilating rate as the free-burning method used. However, if the effect of temperature or humidity upon the perception intensity of tobacco-smoke odor produced by human puffing is different from the effects on tobacco odor produced by the method used in the perception trend studies, then the preceding statement would not apply.

General Discussion of Perception Trends: It had been planned to establish more specific trends in perception intensity; however, the difficulty in obtaining uniformly consistent scores did not make this possible. During the tests, the panel members could make direct comparisons between odor intensities at different relative humidities for the same dry-bulb temperature, but in general no direct comparison was made between the levels at different dry-bulb temperatures. For example, the evaluations made at 80 F dry-bulb were made at least a week after the evaluations made at 75 F or 85 F. Hence the subject had to rely completely on the word description of the sensory scale, since the memory of an exact odor level from the preceding week was next to impossible.

Making judgments on this absolute basis has its disadvantages since fluctuations and deviations in scores are difficult to account for. Using one room as a standard environment and comparing this with the evaluations made in the test room would be a more accurate approach; however, such a method would be more time consuming.

Although the perception areas in this paper show a general trend, more data are needed to determine a locus of points representing constant odor intensities. When these are available, the effect of temperature upon odor perception will be more clearly shown.

ACKNOWLEDGMENTS

The authors wish to express their gratitude to Amos Turk, A. H. Gee, and other members of the Technical Advisory Committee on Odors for their valuable suggestions and guidance throughout the test program.

CONCLUSIONS

1. The intensity level of an odor of constant concentration is lowered by an increase in humidity at constant dry-bulb temperature.
2. The extent of the effect of humidity upon odor perception is not the same for all of the odorants studied.
3. The odor level of cigarette smoke decreases slightly with an increase in dry-bulb temperature at constant specific humidity.
4. Olfactory adaptation occurs more rapidly during the initial stages of exposure.

5. Exposure to cigarette smoke over extended periods of time, while causing olfactory adaptation, will generally have a progressive irritating effect upon the eyes and lining of the nasal passages of the observer.

6. Both sensory scales used in these tests demonstrate that the odor perception level is a function of the logarithm of the concentration.

Since the paper was first published in the JOURNAL SECTION of *Heating, Piping & Air Conditioning*, the application of statistical analysis, using paired and unpaired data for small samples, has substantiated the basic conclusion of the paper, that the intensity level of an odor of constant concentration decreases with an increase in humidity at constant dry-bulb temperature.

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DISCUSSION

AMOS TURK, Danbury, Conn. (WRITTEN): In any investigation in an area of research in which mechanisms are not clearly understood, where truths emerge from statistical analysis of masses of data rather than from a few *crucial* experiments, and where no one is certain as to which experimental factors are critical and which trivial, it is particularly incumbent upon the investigators to exert rigid controls in their work and to report their results in sufficient detail so that workers in other laboratories can check them. In the field of odor measurement, in which the literature is so replete with loosely controlled or incompletely reported studies, it is especially refreshing to read the present paper. When one considers the mass of published data on odor measurements, most of it without specifications of those environmental factors which the present authors have shown to be important, one feels that this paper's major fault is that it was not published a generation ago.

A further indication of the value of an investigation is its ability to stimulate thinking toward new aspects of the research. Specifically, the following possibilities suggest themselves:

1.—Since cigarette smoke is a system containing both vapors and particles, what are the separate roles in odor stimulation of the vapors, the vapors desorbed from the particles, and the particles themselves?

2.—What are the effects of non-odorous nasal irritation on odor perception?

3.—What is the causal nature of the humidity and temperature effects noted: can they be duplicated in studies of animal olfaction; can non-toxic, odorless gases or vapors other than water have a similar effect in odor reduction?

Let us hope that such areas of investigation will be pursued and reported at the level set by the present authors.

WARREN VIESSMAN, Baltimore, Md. (WRITTEN): The authors are to be congratulated on expanding our limited knowledge on the subject of odors. Modern industrial processes and synthetic materials, as well as comfort and health conditions of enclosed spaces make this subject an important one. It also has some military implications.

The fundamental information developed on the effect of temperature and humidity on odor perception is extremely interesting. The results are not necessarily in accordance with one's expectations. As indicated in Figs. 3, 4, 5, and 6, perception characteristics vary somewhat with the vapors under consideration.

A suitable instrument to determine odor intensity is not yet available. Recourse has therefore been made to the human nose as a sensitory instrument. By means of an organoleptic panel, the sensation of a number of trained observers are averaged to give greater accuracy and dependability in the evaluations. The use of a panel instead of a meter has the advantage of providing information on adaptability, or reduced sensitivity over a period of time, of a person exposed to an odor of known concentration. Such a panel is also necessary to determine the irritation to a person when exposed to an odor as a function of time.

The opposite effects of adaptability and irritability are perhaps confusing in the resultant impression. Even with these 2 disturbing phenomena, the authors have been able to develop useful data on the effects of temperature and humidity.

It is hoped that further study and investigation will be forthcoming, which will add to our knowledge of the subject, and be helpful in applications for the reduction and control of odors.

A. B. HUBBARD*, Bloomfield, N. J. (WRITTEN): The authors are to be commended for their diligent pioneering in a difficult field. Their conclusion that odor perception decreases as humidity increases will come as a surprise to many; the opposite being the common impression. Common impressions can be reconciled with this conclusion by postulating that desorption of odors from surfaces increases at a greater rate, with humidity increase, than odor perception decreases. The authors have separated the variables affecting perception from those controlling concentration. They would be the first to agree that their work is only a beginning and to join in the hope that others will be stimulated to extend and clarify the field.

No attempt is made to explain the mechanism of the temperature and humidity effects observed. This is a proper subject for physiologists and psychologists to take up. If one is tempted to adopt the view that water vapor simply masks the odorant, he should first note that Fig. 7 shows that odorant concentration increases tenfold per unit of odorant intensity. Inspection of Figs. 3, 4, and 5 will indicate that odor intensity changes of one point are accompanied by humidity changes far less than tenfold. Therefore, the main mechanism must be the condition of the odor receptors. Perhaps the thickness of a liquid film over the olfactory receptor is greatly influenced by the condition of the air being sniffed.

If there is indeed a physiological reflex, that changes the condition and hence the acuteness of the olfactory receptor, then there must be an adaptation period following a change to new psychrometric conditions. Perhaps it is significant that the 30 percent RH curve in Fig. 10 levels off in about one minute. Since there has been very little change of temperature and humidity experienced by the observers, this curve probably shows the true olfactory fatigue time. The 47 percent RH curve requires 6 minutes to level off. I interpret this to include adaptation to the new humidity.

R. L. KUEHNER and J. D. KARNES†, York, Pa., (WRITTEN): Because the results of this work have major implications not only on the theory of olfaction, but also on odor control practice and, more particularly, on the design of experiments directed toward odor

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† Research Department, York Corporation.

control, we very critically examined this paper, even checking some results experimentally. No major error, or misinterpretation, was found and accordingly we concur with the general conclusions. This is particularly gratifying since we have previously been active opponents of the panel technique for odor measurements.

However, certain qualifying observations are necessary to prevent misuse of the authors' generalizations. The following comments are directed entirely toward the tobacco smoke tests. It must be pointed out that, in the final analysis, the sensory scale numerical values refer to adjectives and, accordingly, are so prone to psychological interpretation that the panel of one organization may give a scale radically different from that of another. Additionally, for the Weber-Fechner Law to hold, the odor sensation at a scale reading of 3 must be physiologically stronger than 2, to the same degree that 2 is than 1, and so on. Evidence that this is true is lacking from the paper.

It is only at scale reading 1, or threshold, that the word description is unqualified by psychological terms and presents a baseline from which comparisons can be made. Therefore, it would be helpful if the authors would record in terms of grams of tobacco burned per cubic foot of air just how much their threshold is. By inference from Fig. 8, we have calculated the threshold level of this panel to be 0.13 mg tobacco burned per cubic foot of air. This appears low to us. Even so, applying this figure and the logarithmic relationship to a 1000 cubic foot office space without ventilation the burning of 1/10 of one cigarette would be barely detectable; one full cigarette will be definite, and with 10 cigarettes the space will still be tolerable, although not pleasant. While an experimental panel can be trained to accept such a concentration as strong, but acceptable, the paying customer in a practical installation is not going to find the odor of anywhere near 10 cigarettes in a 1000 cubic foot space as acceptable, particularly if he is paying for a deodorant. Hence, while the sensory scale may be valid for test purposes, we don't believe it is intended as a practical yardstick for rating odor conditions or deodorant efficiencies. These comments are made to forestall interpretations which we have already seen in the field; that a deodorant must reduce an odor level 10 or 100 times to be practical. We do not feel the authors intend this, or the facts support it.

Additionally, is the data comparing odor levels reached by *human* smoking, mechanical *duct* smoking, and mechanical *in-space* smoking (Fig. 8). Line C on Fig. 8 suggests that more odor is produced by human smoking than mechanical smoking. The authors recognize, and rightly so, that lines C and A are not strictly comparable because of the difference in smoking rate, and suggest that line C should be displaced to the left. When we did this by calculation, it was found that A and C fall practically together. The implication that the same amount of odor is produced by mechanical and human smoking does not seem believable because of the tremendous absorption of odor in the respiratory tract, and clothing of the smokers; evidence which our group has determined experimentally and presented to this society some years ago. Additionally, the discrepancy between *in-duct* and *in-space* smoking is explained by absorption on duct surfaces. This absorption amounts to a 29 percent reduction in odorant concentration. At 200 cfm, it does not seem possible that 29 percent of the odor could be absorbed in one pass through the ducts of this apparatus.

Therefore, another explanation was sought; we believe it to be found in the variation in experimental procedure among the 3 burning methods. In both the in-space mechanical and human smoking, cigarettes were burned for one-half hour before panel determinations were made. We assume smoking continued during the sampling period. With in-duct burning, cigarettes were burned for only 10 minutes before measurements were started, and burning continued for an additional 15 minutes, during which time odor determinations were made. Because of the continuous dilution of a fixed odor source, a certain calculable time must elapse before the odor in the test space reaches equilibrium. With the duct smoking situation, and a ventilation rate of 46 cfm per cigarette (Fig. 6), odor levels reach 90 percent of equilibrium in 9 minutes; 95 percent in 12 minutes, and 99 percent in 19 minutes. In all cases the equilibrium time will vary inversely with the rate of smoking and ventilation. Therefore, the time at which a panel

member took his reading is very critical for in-duct smoking, while in the other 2 methods time is less of a factor. Would not this explain why line A falls below B in Fig. 8, the differences between initial readings on Figs. 10 and 11, and discrepancies of threshold readings at similar conditions on Figs. 6 and 8?

Finally, we want to point out that there is no unexplainable disagreement between this work and ours, the above comments are directed toward getting more information on record that we can make more extensive use of the knowledge. We hope that any criticisms are constructive as they are certainly intended.

C. M. BOSWORTH, Syracuse, N. Y., (WRITTEN): There are numerous articles on odor in the literature indicating that humidity affects the sense of smell. I believe that this paper contains the most data concerning this relationship and is the kind of fundamental data required by the industry. We are indebted to the authors for this data and the analysis.

We all wish that we knew how this water vapor operates on our sense of smell. Those who are interested in ionization of air might say that because the electric charge in the air is changed by humidity change, then odor sensation is an ionization phenomenon. Those who believe that the olfactory membrane works by specific absorption or adsorption might say that the presence of water vapor increases the rate of sorption of the odorous molecule. Possibly it is a surface tension phenomena that is affected by water vapor.

Once the mechanism of odor perception is understood, then we will be able to use possibly electrical, mechanical or chemical measurements for odor analysis rather than the organoleptic panel tests that are now required. These laborious, inexact, and time-consuming panel tests require statistics for analysis. Even with the specially trained panel available to the authors, this paper is a summation of many odor appraisals and if my calculations are correct the number of appraisals are in the order of five thousand! This is a monumental task and the authors deserve a great deal of credit.

H. L. BARNEBEY, Columbus, Ohio (WRITTEN): I should like to comment on the fact that the physical concentration of an odor-causing material must be increased about tenfold to raise the odor intensity by one point. This has great practical significance for the following reasons.

1.—To obtain effective odor control it is necessary to do a rather complete job. For example, if the odor intensity is classified as *strong*, 99 percent of it must be removed to bring the level down to *threshold*. A removal of 90 percent (which might be thought of as a high figure in some types of work) would be relatively ineffective because the intensity would have been brought down only one point or to *definite*.

2.—With activated charcoal adsorption equipment, it is possible to economically achieve the high efficiencies of removal required for effective odor control. It requires little additional equipment or causes little additional loss of activated charcoal's capacity to achieve 99 percent removal as compared with 90 percent.

3.—The human nose is a poor judge of the amount of odor to be removed and should not be used as the basis for designing a ventilating or odor removal system. An error of one point will constitute a tenfold mistake in the amount of work to be done. The concentrations of odors should be determined by analytical means or a portion of the space purified or ventilated by the contemplated means to establish the type and size of equipment.

A. HALDANE GEE*, New York, N. Y.: Having attempted some of the analyses the discussers went through I feel that the authors and the discussers are certainly to be congratulated.

A point one of the discussers raised is the relative effect of the vapors from the cigarette smoke or of the particles or of the mist. The smoke may contain gas, or aerosolized

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liquid, or suspended solid, so it would seem worthwhile to electrostatically separate the visible material and see what the ratings then are.

Since we have a nice system set up for determining the odor levels, this would be an interesting further outcome. It would have utility in analyzing further the question as to what extent perceptible odors are in the form of vapors or gases and to what extent they can be in the form of liquid or solid particles. This has a definite application in odor control by aerosol methods, for which a great many cans of aerosol are being sold—whether the liquid particles remain suspended or whether they have any effect as particles.

I have a couple of other comments. I do feel that the authors have well supported the view that the increases in perception levels in this scale are logarithmic. The comparison we use from time to time, on the 4-point scale, is that zero is something that weighs nothing at all, one is featherweight, two is a handful, three an armful, on up to four which would be something you couldn't lift at all, such as a large table. We remind our own panelists of this from time to time, which helps to maintain uniformity in evaluations.

Just one other minor point on the question of the density of their smoke. The authors probably mentioned in their paper just about how many cigarettes in what size of room this would be equivalent to. Now it would be interesting to know from them what the test smoke levels were like. Did they cause irritation or not, whether they saw the smoke or not. Just what was visible, in terms of a level of concentration—a light living-room haze, or the haze of a *smoke-filled back room* or was the test smoke at exaggerated levels in terms of occupied space?

T. H. URDAHL, Washington, D. C.: Most of you have seen me up here before at meetings where there has been a discussion dealing with odors. I have this year been relieved of the Chairmanship of this Committee, which I have enjoyed, and which has so far resulted in the production of this paper.

While our authors have, so far most deservedly been highly complimented, I cannot too highly compliment the energy and zeal with which they have approached and brought out these results in a most nebulous field.

In the initial approach to this problem, it was realized I think by all who were on the committee, as well as many from the Society, that we were embarking on something highly intangible, many facets of which were way beyond the engineering field in which we work. Our interests are primarily in getting answers because answers are important to the solution of our engineering problems.

We have, accordingly and to some extent—and we should increase the extent—acted as a catalyst upon the other professions whose field of investigation would get further into the mechanisms of the human fact, which as you have heard, is one of the roots of this problem, mechanical smelling devices not withstanding.

One of my interests in odors stems from experience, now many years ago in the United States Navy and many of you have heard my references to the work of the Navy.

The problem to be solved was not that of odors because odors were not considered to have a significant bearing on the comfort or well-being or health of a seaman. However, quantities of so-called freshener were important, and toward the latter years, in fact the work was begun when World War II was ended, the physiological and medical components of the Navy were requested to conduct a mass scale experiment to determine the minimal quantity of replenishment of fresh air required for comfort at a given shipboard air condition. In other words, one in which cooling coils were used to maintain an effective temperature of 78 F. The results of this experiment are contained in the publication now unclassified, and in checking the results which our authors here have gotten, against some of the results, you might say impressions, of this Navy mass scale research, I find it most interesting to note that these results checked out very closely.

The Navy's test did not resemble pure research to the extent reflected in the work done under the Society's auspices. It was an attempt to obtain an answer under actual con-

ditions in the shortest possible time, and also to evaluate as a side issue the efficacy of known odor controlling elements such as activated carbon and a number of other items.

This experiment, conducted over a period of one year, used the largest number of people which I believe has ever been used in history. There were continually, at least 46 people normally living within the space simulating that aboard a ship, conducting themselves in the normal manner of life at sea, as nearly as possible to simulate at best, these conditions at the Bethesda Naval Hospital, on dry land.

The smoking lamp was always lit. We had the influence of cigarette smoke, possible cigar smoke, and many of the conclusions which have been noted in this paper throw light upon some of the results obtained in the Navy's work. In other words, as I have said, the results of this mass experiment were not interpreted. It just occurred and the conclusions which are shown here are listed from 1 to 6 are well substantiated by this actual mass scale experiment which was conducted some 11 years ago.

H. B. NOTTAGE, Encino, Calif.: What about a possible influence of microscopic dust particles? Was the air always uniformly *clean*? Some odor gases might be chemically or physically active in contact with dust particles, and this could be a disturbing influence.

Also, an important question is clothing as a possible variable effect. Did the authors make certain that all panel participants had the same outer garments, uniformly *clean*, for each test?

AUTHORS' CLOSURE (Mr. Kerka): In regard to Mr. Viessman's statement concerning the basic finding of the paper, viz: that the odor perception level decreases with an increase in humidity seems contrary to one's expectations, we cannot overemphasize the fact that in each perception-trend study the physical concentration of odorant remained constant. In actual environments, no such controlled conditions exist; and the rate of odorant production as affected by temperature and humidity, together with degree of air motion, will vary odorant concentration. There is no doubt that the increased amount of odorant concentration under humid conditions causes one to experience higher perception levels. Some of these points have been brought out in Dr. Kuehner's paper presented in Cincinnati last January.

The fact that adaptation to an odor as opposed to irritability may appear confusing is probably due to the fact that the average person does not separate these 2 effects, but lumps them into one impression. It is possible for some substances to be odorous without being irritable and vice versa. In the case of tobacco smoke, both components are present; and hence separate considerations should be made. Until a suitable instrument is developed that will measure the responses of the olfactory system, the human nose must serve as our primary judge, subject to physiological and psychological fluctuations as it may be.

Mr. Hubbard is certainly correct in stating that this paper is only a beginning. The phases of study in olfaction, odor perception, and the physical behavior of odorants are multitudinous. It is hoped that the basic findings of this paper will serve as a guidepost for future investigators, and will bring home the realization that environmental conditions must be closely controlled whenever any organoleptic techniques are employed. Since the exact mechanism of olfaction is not fully known, any conclusion drawn from the results of the paper with respect to the physiological response of the receptors can be only speculatory in nature.

The curves in Fig. 10 were presented to show only general trends. It is probably not justifiable to draw definite conclusions from individual points along any curve.

Some very important points have been brought up by Dr. R. L. Kuehner and Mr. J. D. Karnes regarding the application of the sensory scale to cigarette smoke. Without a doubt the sensory scales are far from being absolute and only the threshold level has any true physical significance that can be accurately measured. Reference is made to Fig. 8 and the comparison of burning 1/10, 1, and 10 cigarettes in a 1,000 cubic foot space. It is true that the panel members rated these three conditions as noted in the

discussion. Undoubtedly one would not tolerate this latter condition in a bedroom or office space; but one would possibly be more resigned to it in a *smoke-filled card room* or pool hall where such conditions are expected.

The panel members could through training accept such a concentration as *strong* but *endurable* only on a first-impression basis, however. The results of Fig. 10 indicate that continuous exposure to such a condition as 10 cigarettes in a 1,000 cubic foot space could result in an intolerable irritation level, as would be expected. Hence, it is important, especially in evaluating cigarette smoke, to realize whether it is odor level or irritation level that is being judged, and whether these are first impressions, or impressions made after periods of exposure. Since some misinterpretation of the sensory scales can result, it could be advantageous to use a more absolute means for rating odor conditions or deodorant efficiencies.

It is with some regret that the results for the burning rates of cigarettes were not reported on a weight burned per cubic foot basis. The units used, that is, cfm per cigarette with reference to the burning rate, were felt to be practical in nature and hence were adopted. A calculation based on weight burned per cubic foot shows line C of Fig. 8 to be nearly coincidental with line B rather than with line A as pointed out in the discussion. The methods of smoke introduction represented by C and B are more comparable since the smoke was generated in the test rooms in both cases. By this comparison, line C should fall below line B, since, as Dr. Kuehner and Mr. Karnes have pointed out, absorption in the respiratory system of the smoker reduces the output of odor. The fact that the evaluations did not comply with this may indicate a psychological response from the panel members. The awareness of human smokers puffing in the rooms could have accounted for a higher-than-normal rating.

Adsorption on the duct surfaces is still felt to be at least partially responsible for the differences between methods A and B in Fig. 8. Four steel mixing vanes located after the point of odor introduction would have contributed to this. As in the case of the human smokers, being aware of the cigarettes in the room may have resulted in overevaluating the condition. It was pointed out that the in-duct method reached only 90 percent equilibrium after 9 minutes when evaluations were begun and that this equilibrium reached 99 percent near the end of the run, hence making the time of judgment critical. It had been found that the panel members could not normally detect any change in odorant concentration less than 33 percent, and in some cases 50 percent changes were difficult to detect. Therefore, it would have been improbable for them to detect any difference between 90 and 99 percent equilibrium. The method of smoke introduction used for Fig. 6 is represented by line A of Fig. 8; and any discrepancy of threshold reading is of the order of only 0.3 of a point on the sensory scale used.

Dr. Kuehner's and Mr. Karnes' criticisms bring out some of the problems of using the organoleptic panel.

Both Mr. Barnebey and Dr. Kuehner indicate that means other than the use of the organoleptic panel should be employed for odor-control work. It is agreed that the use of physical analysis of odorant concentration and dilution techniques as suggested are more absolute methods.

In answering Dr. Nottage's question, I can only say that every effort was made to purify the inlet air to the test rooms through a series of mechanical, electrostatic, activated carbon, and absolute filters. Contaminated dust particles could have entered the test rooms from the outer laboratory (during entering and leaving the rooms), but any odor adsorbed on their surface would have been minute and negligible to the test odor being measured. The question of wearing a special garment for the tests came early in the development of the program. At that time, it was decided that wearing a uniform or cover of low odor-adsorbing qualities would be deviating from the natural experience of walking into a room normally clothed. The women of the panel were requested not to use excessive perfume. Again these minor contaminations were below the odor level being judged.

In answer to Dr. Cee, the condition in the test rooms during the perception-trend studies was equivalent to about 5 cigarettes burning in a 1,000 cubic foot space with a

ventilation rate of 46 cfm per cigarette. When the smoke was introduced into the air-supply duct, no visible haze was noted in the test room, although irritation was noted with time of exposure, especially under low relative humidities as shown in Fig. 11. With smoke generated in the test space itself, either by free-burning or human-puffing action, visible hazes were detected. At the low ventilating rates as shown in Fig. 8, a haze was visible in the room, but its exact density was not analyzed. No evaluations were made to record the rate at which the haze cleared as the ventilation was increased.

1588

No. 1588

THE INDIRECT EVAPORATIVE AIR COOLER

By J. R. WATT* AND R. A. BACON**, AUSTIN, TEX.

TWENTY YEARS ago, thousands of Arizona and California business establishments and homes were cooled by air conditioners using evaporating water to chill air without humidifying it. Largely forgotten today, these indirect evaporative coolers usually had either finned-coil or plate-type heat exchangers.

A finned-coil cooling system comprised an oversize natural-draft cooling tower, a pump, a fan delivering air indoors, and a copper radiator for cooling it. Automobile radiators served until specialized types appeared.

Given low local humidity, the cooling towers (about 4 x 8 ft x 20 ft high for residences) often delivered water 35 deg below outside dry-bulb temperature or within 3 deg of outside wet-bulb temperature. The water was pumped through radiators having from 4 to 8 rows of tubes, and counter-flow to air drawn through by the fan. Since the air often emerged within 1 deg of entering water temperature, many systems delivered it only 4 deg above outside wet-bulb temperature, or frequently 30 deg or more below outside dry-bulb temperature.

The most successful installation was probably the regenerative system in the University of Arizona Administration Building. This operated continuously from 1936 until 1952, when it was supplanted by campus-wide refrigeration. The original cooling tower, built into the two-story building, was a vertical shaft of 4 x 5-ft cross section extending from basement to attic. About 135 gpm of water sprayed down through 16,000 cfm of air, largely made up of cool returns from the rooms, rising upward. The counterflow, multirow radiator then cooled 10,800 cfm of filtered outside air which was ducted throughout the building. The cool and humid tower air escaped through the attic to dissipate roof heat.

Delivered air temperatures were unusually low, because the regenerative return air reduced tower wet-bulb temperatures below those outside, allowing lower water temperatures and increased output. Indoor conditions thus averaged at least 15 deg below outside temperature.

The estimated 42 tons of cooling required only 17½ hp, plus water consumption comparable to corresponding refrigeration systems. The only moving parts were 2 centrifugal fans and a pump, aside from 10 tons of supplementary refrigeration used in humid weather.

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This cooling system received only filtered air and softened water plus a weekly tower flushing. Many Arizona installations accumulated air- and water-borne minerals in the water until scale lined the radiator tubes and reduced efficiency. Seasonal radiator cleanings caused frequent replacement until wartime shortages intervened.

Many systems survived in California, and installations are made there today. Redwood towers with water-treatment and monthly contract cleaning are combined with multi-pass radiators with removable headers. With air and water velocities around 600 and 200 fpm, respectively, such coils cool air virtually to entering water temperature. Occasional towers are two-stage, the better to approach wet-bulb temperature. Many systems serve as complete cooling plants; others merely pre-cool for refrigerated air conditioning.

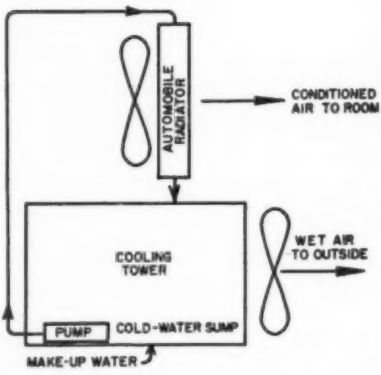

FIG. 1—INDIRECT EVAPORATIVE COOLER WITH FINNED COIL

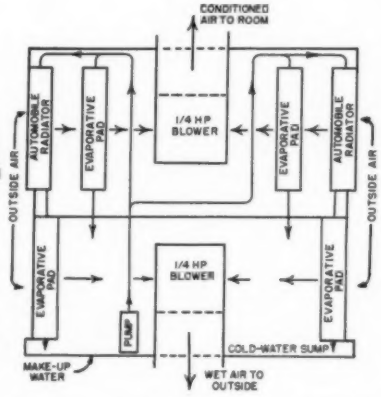

FIG. 2—TWO-STAGE EVAPORATIVE COOLER. CONDITIONED AIR INDIRECTLY COOLED AND THEN EVAPORATIVELY COOLED

Some systems, on the other hand, have direct-evaporative second stages, the dry-cooled air being further chilled by adiabatic humidification in air washers or saturated pads. Since indirect cooling lowers the wet-bulb temperature of air, the direct second-stage often delivers air several degrees below outside wet-bulb temperature. Such cooling is necessarily used like single-stage direct-evaporative cooling. As delivered air frequently exceeds 70 percent relative humidity, all fresh air is used without recirculation.

The plate-type indirect evaporative coolers originated in Los Angeles. About 1930, massive plate-type air-to-air heat transfer units were installed to cool several large buildings. More compact machines were installed widely in chain store buildings.

These coolers consisted of multiple, hollow, galvanized plates, 2 ft wide, 5 ft tall and $\frac{5}{16}$ in. thick. Spaced $\frac{1}{4}$ in. apart, these formed narrow parallel flues separated by thin metal. One fan drew filtered outside air and fine water spray

down between the plates, cooling them by evaporation. Another fan drew filtered outside air upward inside the plates and delivered it to the building. The water was filtered to protect the spray-nozzles, and the wet air was discharged direct to outdoors or through attic space.

These systems gave economical cooling with all fresh air. One that was installed in Tucson in 1936 had 20,000 sq ft of cooling surface to produce about 60 tons of cooling effect from 19½ hp. It was still operating in 1955 and required little attention other than occasional servicing of clogged nozzles and rusted wet-air fan blades.

Aside from the traditional types, new designs have appeared. In one Texas residential system, 4 automobile radiators connected in a closed water circuit formed

TABLE 1—PERFORMANCE OF 2-STAGE FINNED-COIL COOLER

	RUN 1	RUN 2
Volume of conditioned air, cfm	2000	2000
Power supplied, hp	5/6	5/6
Outside air db temp. F	99.1	88.2
Outside air wb temp. F	76.5	74.4
Outside air relative humidity, %	37	52
Air from cooling coils, db temp. F	86.7	80.5
Air from cooling coils, wb temp. F	73.3	71.8
Air from cooling coils, relative humidity, %	53	67
Heat removed by cooling coils, Btu per min	420	260
Heat removed by cooling coils, tons per hp	2.5	1.6
Cooling effectiveness of cooling coils, %	55	56
Cooling rate of cooling coils, Btu per (min) (F deg of wb depression)	19.3	19.6
Air from second stage pads, db temp. F	73	72.3
Air from second stage pads, wb temp. F	72.5	71.8
Air from second stage pads, relative humidity, %	96	96
Conditioned air total db temp. decrease, F deg	26.1	15.9
Conditioned air total sensible heat removed, Btu per min	760	465
Conditioned air total sensible heat removed, tons per hp.	4.60	2.80

tower and cooling coil. One fan drew make-up and return air, plus a city-water spray, over 2 radiators and discharged them. The closed-circuit water, cooled by evaporation in these 2 radiators, flowed to the other 2 where it chilled the supply air for the rooms. Under test, 5/6 hp cooled 3700 cfm of air to 10 deg above ambient wet bulb. The spray water was discharged onto the flat roof.

One type of indirect evaporative cooler resembles regenerative air preheaters used in power plants. A porous heat-transfer wheel packed with aluminum wool revolves 20 to 30 rpm. It is supplied with evaporatively cooled return air through part of its area. This air chills the aluminum wool to below outside wet-bulb temperature, and the wool then revolves into the path of filtered, dry outside air flowing to the rooms. After cooling the room air to near wet-bulb level, the packing revolves back for rechilling.

EXPERIMENTS IN INDIRECT EVAPORATIVE COOLING

In 1952 and 1953 the University of Texas conducted research upon indirect evaporative coolers. Five types were tested, including 2 finned-coil and 3 plate-type units, one of each being two-stage.

One finned-coil system with an oversize horizontal forced-draft redwood cooling tower, illustrated in Fig. 1, circulated water at only 1 deg above outside wet-bulb temperature through a new fan-equipped automobile radiator. Tests showed

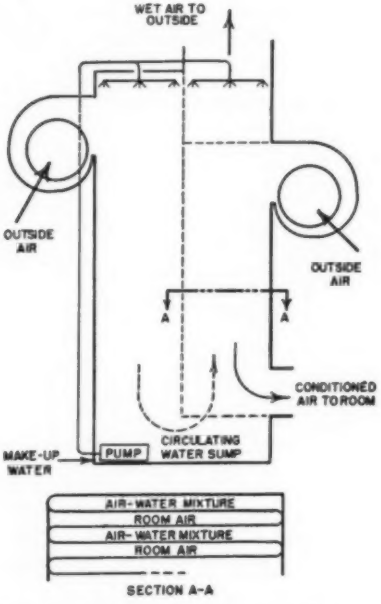

FIG. 3—INDIRECT EVAPORATIVE COOLER WITH FOLDED PLATE-TYPE HEAT EXCHANGER

that 11.8 gpm of water flowing through the single-pass radiator cooled 1490 cfm of air to 5.5 deg above outside wet-bulb temperature. The air was cooled 6.3 deg and the wet-bulb depression was 11.8 deg, which gave a cooling effectiveness‡ of 54 percent.

Lesser water rates resulted in warmer radiator water and warmer air. Flow of 3.8 gpm gave air 9.3 deg greater than outside wet-bulb temperature and a cooling effectiveness of 46 percent, the air rate being 1490 cfm as before. The cooling rate was 11.6 Btu per (min) (F deg of wet-bulb depression), whereas a water rate of 11.8 gpm gave 13.4 Btu per (min) (F deg of wet-bulb depression).

‡ Cooling effectiveness is the ratio of dry-bulb temperature decrease to outside-air wet-bulb depression.

The second finned-coil machine, illustrated in Fig. 2, was two-stage. Outside air drawn through the automobile-type radiators was cooled by heat removal and then passed through wet aspenwood pads, which formed the second stage, in which it was adiabatically cooled. The lower chamber formed an induced draft cooling tower. Enclosed by aspenwood pads kept wet by recirculating water, the lower blower drew air through them and discharged it below. Water circulated from

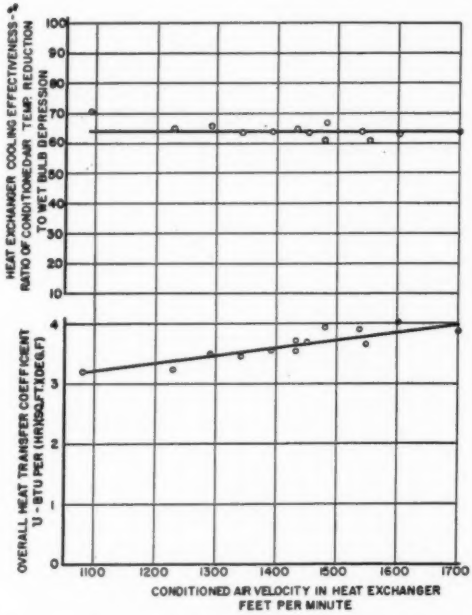

FIG. 4—COOLING EFFECTIVENESS AND OVERALL COEFFICIENT OF HEAT TRANSFER AT VARIOUS VELOCITIES IN THE INDIRECT EVAPORATIVE COOLER ILLUSTRATED IN FIG. 3

the sump through the radiators, with some by-passed to the second stage pads, and then through the cooling-tower pads. The performance of this machine is shown in Table 1.

The dry-bulb temperatures of the conditioned air delivered were below outside air wet-bulb temperatures in both runs. They were below the 80 F dry-bulb temperature commonly needed for comfort, and, hence, there might have been less cooling in the pads, with the air delivered still being at satisfactory temperature and at lower relative humidity.

Heat removed in the cooling coils was over half of the sensible heat removed. Cooling effectiveness of 100 percent in the coils would realize the full potential of cooling that could be achieved by use of the pads alone and with the added advantage that the conditioned air would have less humidity.

The first plate-type cooler, illustrated in Fig. 3, was composed of a vertical casing enclosing on the left a spray chamber and on the right a sheet-aluminum heat exchanger whose metal folds formed multiple hollow plates spaced ⅜ in. apart. These spaces provided a continuation of the spray chamber wherein falling water sprays

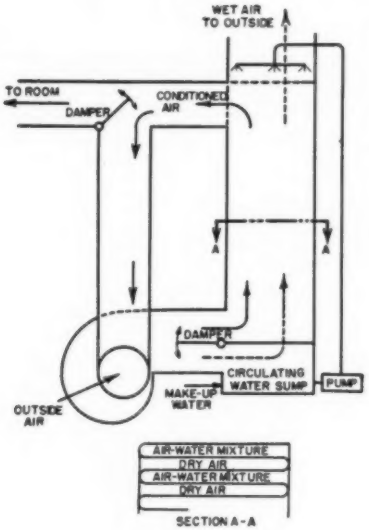

FIG. 5—REGENERATIVE INDIRECT EVAPORATIVE COOLER WITH FOLDED PLATE-TYPE HEAT EXCHANGER

mixed with rising air and absorbed heat from the dry air through the metal that it continuously wetted. The dry air and the air through the spray chamber were driven by separate blowers. The sprays in the chamber on the left proved needless, so tests were conducted without them.

This machine's overall heat transfer coefficient varied between 3 and 4 Btu per (hr) (sq ft) (F deg), as shown by the curve in Fig. 4, and the dry air was cooled to over 60 percent of the prevailing wet-bulb depression.

The second plate-type indirect evaporative cooler, illustrated in Fig. 5, was similar to the one illustrated in Fig. 3. A housing enclosed a corresponding vertical heat exchanger with hollow plates ¼ in. thick and spaced ¼ in. apart. One blower supplied both wet-side and dry-side air, and a damper provided a control for the division of the discharge from it. Air flow was upward in both sides, and water sprayed downward in the wet side. A damper at the dry-air discharge provided

a control for recycling any part of the air from it into the blower to test for regeneration effects.

No appreciable regeneration gain was achieved. When—the outside wet-bulb depression being 20 deg—10 percent of the dry air was recycled, it was delivered no cooler than when none of it was recycled. Larger proportions being recycled gave no measured improvement, and, of course, recycling reduced the amount of air discharged for comfort cooling. The blower discharged 60 percent of its air into the wet side and 40 percent into the dry side. Recycling 10 percent of the dry air provided only 4 percent of the air for the blower, the other 96 percent being outside air; recycling 100 percent of the dry air provided only 40 percent of the air for the blower and produced no air for comfort cooling. Regeneration by using cool return room air through the wet side is advantageous, however, for this commonly provides the blower with air having lower wet-bulb temperature than recycling

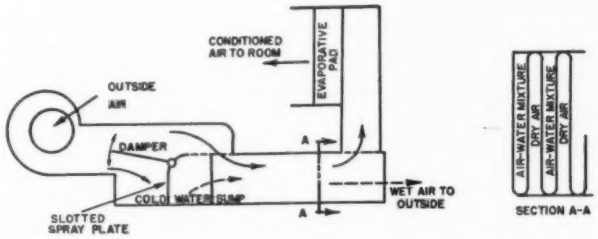

FIG. 6—HORIZONTAL TWO-STAGE EVAPORATIVE COOLER. SLOTTED PLATE GIVES AIR-BLOWN WATER SPRAY

does. Overall heat transfer coefficients were found to be from 4 to 6 Btu per (hr) (sq ft) (F deg), and cooling effectiveness was between 60 and 70 percent.

The third plate-type cooler, illustrated in Fig. 6, had a heat exchanger that was horizontal, and one blower supplied both wet-side and dry-side air. The slotted plate resembled a comb; all wet-side air passed through the slots and sprayed sump water into the heat exchanger. The cooler was equipped with a second stage for evaporative cooling.

The cooling effectiveness of the heat exchanger was between 40 and 50 percent. Water droplets did not carry well enough to wet the heat exchanger near the discharge end. The direct evaporative cooling stage cooled the air to between 2 deg below and 1 deg above ambient wet-bulb temperature.

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DISCUSSION

S. F. DUNCAN, Torrance, Calif., (WRITTEN): The paper shows that various arrangements of equipment can be used to derive cool air from indirect evaporative cooling. The fact that no pertinent dimensions or type descriptions are given makes it impossible to evaluate the equipment from the standpoint of practicability. Anyone wishing to take advantage of the performance figures given would be unable to design similar equipment for lack of dimensions and similar data. The paper would be much more useful if, for instance, such data as the area and thickness of the aspen pads in Fig. 2, had been given.

W. T. SMITH, Washington, D. C.: I would like to take this opportunity to express appreciation to people like Professor Watt. Evaporative cooling in the Armed Forces represents an economical and highly practical means of relieving the intense heat that exists in many parts of the world.

I can recall in San Francisco just a year ago I made the appeal to industry to please study this problem and study it thoroughly. I would like to express appreciation to Professor Watt for bringing to your attention the fact that you can bring air down below the wet temperature by means of what appears to be an aeromatic apparatus.

We have a gentleman here from the Air Research and Development Command and I believe we do have in operation a two-stage evaporative cooling system on the building. It is unfortunate that I did not get this paper in time or I would have been glad to tell you something about that.

Now, in final comment, what worries me is that if we put too much money—let me impress you with this—if we put too much money into an evaporative cooling installation, we become competitors to air conditioning and the justification for evaporative cooling vanishes.

WARREN VIESSMAN, Baltimore, Md.: While associated with the U. S. Naval Civil Engineering Research and Evaluation Laboratory, at Port Hueneme, California, I became aware of Dr. Watt's interest in evaporative coolers. These coolers are extensively used in arid environments. Since the military services have many installations in dry climates, it was considered desirable to investigate the use and application of this equipment and its possibility of further improvement. The interest and ability of Dr. Watt and his associates offered the opportunity to make such an investigation to further the development of the art and its application, and to determine the limitations of the equipment.

What you have seen and heard here today are to some extent a part of that investigation. The authors have developed some very interesting and unusual things. They have shown that the conditioned air can be cooled without refrigeration to a dry bulb temperature below that of the outside air wet bulb temperature. This may look like

it is possible to go down to most any temperature by running the air around and around through heat exchangers of some sort. However this is not perpetual motion in some new form. The air is never cooled by the equipment to a dry bulb temperature below its wet bulb temperature.

There is a theoretical and economic bottom that is soon reached. Economic reasons, first cost and maintenance costs limit considerably what can be done in practice.

The advantage of using a water cooling coil with its cooling tower in connection with an evaporative cooler, is that the air can be cooled to a lower temperature requiring less air to be provided for the heat load. This greatly reduces the quantity of air to be handled, and the duct and equipment size. The quantity of air required will be more nearly that of a conventional hot air heating system. This makes it possible to utilize smaller ducts and to design for a summer-winter combined cooling and heating system.

The authors have shown what can be done with this very effective and economical means of cooling in arid areas. The historical development of the subject in which Mr. Arthur Hess played a part, is of considerable interest and value. For those here in the East who have never experienced evaporative cooling, I might say that last week I returned from Nevada where the temperature in the afternoon was as high as 107 F. It was very comfortable with evaporative cooling, the high humidity air seemed to add to the comfort in the dry desert country. In California and Nevada, I saw many first class installations of combined summer 2-stage cooling and winter gas-fired heating systems, using air as the heat transfer medium in the same duct work. Some of these installations are not small. Several were very large. Their performance has been enthusiastically received by the people who experience their comfort, and also by the maintenance people involved.

Evaporative cooling does play a very important role in cooling economy, as the authors have so ably shown. Their calculations indicate 2.8 to 4.6 tons of cooling per horsepower, as compared with approximately 1 ton per horsepower obtained with mechanical refrigerant air conditioning systems. This is for 2-stage cooling.

In the event hard or corrosive water is all that is available, as is frequently the case in many desert areas, the water should be treated and the systems kept as simple as possible, in order that maintenance may be at a minimum.

JACK KICE, Wichita, Kans.: I happen to be from a part of the country where this type of cooling is very practical. I was keenly interested in the paper and its practical possibilities seem more exciting to me than the more extreme ideas such as cooling by electronics and atomic power.

Perhaps some of the developments in the near future could be directed to practical application of these principles that we are all acquainted with and haven't used to the fullest extent.

I have just a question or two. One question is whether the impression I have is correct that the theoretical limit of air temperature drop with multiple stage evaporative cooling would be the dew-point temperature of the air regardless of the number of stages. The other question is what decision has been made concerning the most practical type of fan or blower to be used in moving the air. Since moving air seems to be the main work to be done here, the most efficient and most effective type of air moving equipment should certainly be used.

Also, I wonder if other types of evaporative media other than the aspen wood excelsior have been used. If so—what has been the experience?

AUTHOR'S CLOSURE (Professor Watt): Mr. Duncan asked what data were available for designing actual indirect evaporative coolers.

Such information is extremely scattered. Professor Bacon and I, at Texas, developed complete thermal and heat transfer equations for machines with plate-type heat exchangers. These were included in *Investigation of Evaporative Air Cooling*, our report to the U. S. Naval Civil Engineering Research and Evaluation Laboratory, Port Hueneme, California. These will appear in a paper awaiting current tests on optimum

ratios of air to water and wet or *tower* air to conditioned air. Earlier experiments suggest equal weights of *tower* water and air, the latter averaging about 50 percent over the air being conditioned.

Charles Hollingsworth of Los Angeles is an authority on systems using cooling towers and dry-surface coils. Neal Pennington of Tucson invented and knows indirect evaporative coolers using thermal storage wheels. Professor Gafford of Texas developed one with transfer by circulating water, which Professor Hartwig and I are attempting to improve.

Mr. Smith noted the Armed Forces' great need for some form of cooling where budgets prohibit refrigeration. They need cooling to keep men happy and productive, but can not afford either investment or power cost of refrigeration. Besides this, America has perhaps 600 million square feet of uncooled offices and 20 times that of uncooled factory space. Probably one-third of this needs a junior grade of air conditioning, judging by falling productivity as summer temperatures rise.

I think indirect evaporative cooling, delivering either dry air or slightly humidified air, may solve such needs, saving possibly 25 percent on equipment and probably 40 percent on power.

Mr. Kice asked how low successive stages of indirect evaporative cooling can cool air. It seems most economical to stage these machines either with an indirect stage preceding a direct stage, a combination we found very economical, or an indirect stage followed by a refrigeration stage, an arrangement operating well in California. I suspect successive stages of indirect cooling might prove uneconomical due to bulk of fans and heat transfer units, but theoretically enough stages could cool air to its dew-point.

Thus far, cooling air below outside air wet-bulb temperature in one indirect stage is possible only when the *tower* side operates on cool return air from the conditioned space. An indirect plus a direct stage achieve it easily with all outside air.

Concerning fans, the flow resistance of heat transfer units is significant. Plate-type heat exchangers have long air passages and high resistance so centrifugal fans are needed. Ductless systems with finned coils can use propeller fans, however.

Concerning filler for direct stage saturation pads, aspen wood excelsior wets the air best, followed by expanded paper mats. Glass fibres wet less well, but do not need seasonal replacement.

Mr. Viessman emphasized the Armed Forces' need for inexpensive cooling. I agree with his comments on advantages and costs and on water problems. Scale from mineral-laden water greatly reduces heat-transfer in coils if towers are not cleaned and water treated.

Professor Nottage asked about general range of climate required for indirect evaporative cooling. As noted before, these machines cool air in direct proportion to prevailing wet-bulb depression. Indirect stages reduce air temperature 60–70 percent of wet-bulb depression, direct stages 80–85 percent.

Specifying climatic areas suitable for indirect evaporative coolers presents an economic question. In general they should be suitable wherever summer afternoon wet-bulb temperature averages 20 degrees below the dry-bulb.

1589

No. 1589

FUTURE ELECTRONIC, SOLAR AND NUCLEAR DEVELOPMENTS IN ALL-YEAR RESIDENTIAL AIR CONDITIONING

By W. F. FRIEND*, NEW YORK, N. Y.

A CHARACTERISTIC of all 3 of the specific developments named in the title is that, technologically, they still are at the early stages of research and development. They require *basic research*, which means *critical and exhaustive investigation and experimentation with the aim of discovering new facts and correctly interpreting them.* This research must further be implemented by *development*, whereby the *fruits of discovery become more available or useful, through a succession of stages each preparatory to the next.*

Moreover, in a broad way each of the subjects is related to power and energy—their production and utilization, and thus with conservation of world resources. Unless a continuing and expanding supply of energy, both thermal and electrical, can be assured through the years ahead, the material advances that have been gaining momentum through the past half-century will ultimately be checked or even reversed. We hear that residential energy consumption per customer, in the United States now averaging about 2500 kwhr annually, can be expected to reach a level between 10,000 and 20,000 within the next 2 decades. Air conditioning is one of the giant strides toward better living, and already it has accounted for a major component of the increasing load from residential users, who must be served by the electric and gas utility systems of the nation.

Growth of population and changing economic trends likewise are important factors in the expansion of electric utility systems over the country as a whole. Since the last World War they have seen their loads double about every 10 years, and recently there is talk of a doubling in as little as 8 years in some regions. This strongly supports the conclusion that energy resources must be augmented soon, by bringing solar and nuclear power to the stage of commercial feasibility, lest conventional fuels under the pressure of rising consumption become prohibitive in cost.

The necessity for extensive research, before the principles of electronic refrigeration and of solar and nuclear energy conversion can be defined and their utilization reduced to practice, should not discourage engineers belonging to this Society. They were pioneers in establishing broad research activities within their profession,

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Presented at a Symposium on Residential Air Conditioning at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND AIR-CONDITIONING ENGINEERS, Washington, June 1956, under the title, *Future Developments—Electronic, Solar, Nuclear*; see also abstract on pages 137 to 139, JOURNAL SECTION, *Heating, Piping & Air Conditioning*, September 1956.

and in founding and operating a research laboratory of world-wide reputation. The work conducted by the Society on the fundamentals of maintaining human comfort and health through control of the 5 basic elements that compose air conditioning, and the perfecting of equipment types that embody those principles, have been outstanding achievements extending through nearly 3 decades. Results equally successful can confidently be expected in these new domains, within a comparable period.

ELECTRONIC COOLING

By this term is meant refrigeration, or removal of heat, the temperature reduction being produced directly by the flow of an electric current—in this case the movement of electrons within a continuous circuit composed of solid materials. The use of socalled *electronic* devices and complex circuits like those employed in

TABLE 1— EFFECTS PRESENT IN ELECTRONIC COOLING

EFFECT	DEVICE	CAUSE	RESULT
Seebeck	Junction of dissimilar metals	Temperature difference	Electric current
Peltier	Junction of dissimilar metals	Electric current	Heat emission or absorption
Thomson	Single conductor	Temp gradient; also current	Heat emission or absorption
Joule	Electric circuit	Electric current	Heat generation and emission

radio and television apparatus is not involved. The principles in general are called *thermoelectric effects*, a term denoting energy conversion as between heat and electricity.

Four effects, (Table 1) identified by the names of their respective discoverers, are present in electronic cooling. Likewise involved is the principle of conduction, wherein heat is transported or an electric current made to flow between 2 points, by creating a difference of temperature or electric potential. The several effects are stated in the phrasing conventionally employed, in Table 2; reversibility of the action is indicated by entries in the column at the right.

Usually the Peltier effect, which can be considered the converse of the Seebeck effect utilized in the ordinary thermocouple for temperature measurement by the electrical method, is said to be the essential principle of electronic refrigeration. However, each of the other principles is also involved in such apparatus and must be taken into account in its design.

For a successful device, the selection of materials is of primary importance. Three criteria must be satisfied as listed in Table 3. Thermoelectric power (a) is defined as the rate of change of thermoelectric force (or voltage) with temperature.

Until recently, no materials were known, possessing properties that could satisfy the conflicting requirements well enough to enable building a device with acceptable performance. The materials principally used by early experimenters in electronic cooling were antimony, bismuth, copper, germanium, silicon, tin and zinc. They proved unsatisfactory, since the maximum cooling ranges obtainable varied only between 9 and 16 C (15 and 30 F), and efficiencies were extremely low.

Applicable materials are now generally classified under 3 types: pure metals, alloys, semiconductors. Characteristics of these with respect to the performance criteria are as listed in Table 4, the reference letters being those previously defined. Thermoelectric powers for semiconductors may range up to hundreds of times those for metals. The conductivity ratio (thermal-to-electrical), known as the Wiedermann-Franz ratio, is approximately constant for all pure metals, with value 2.3 x 10^{-8} expressed in appropriate units.

Thus semiconductors offer real promise and are being closely investigated. Their unique properties arise from the type and concentration of impurities, present either

TABLE 2—THERMOELECTRIC REFRIGERATION FOR AIR CONDITIONING—BASIC PRINCIPLES AND DEFINITIONS

EFFECT	ACTION PRODUCED	CONDITION MAINTAINED	REVERSIBILITY
1—Thermoelectric	Energy interchange, or conversion between heat and electricity	Difference of temperature or of electrical potential, between two points or across a junction	See (2) to (5)
2—Seebeck (1821)	Electric current within a closed circuit composed of two dissimilar conducting metals	Junctions kept at different temperatures	Yes, then becoming (3)
3—Peltier (1834)	Heat absorption or emission produced at junction of two dissimilar conducting metals	Electric current passed through junction by application of voltage	Yes, then becoming (2)
	Direction of current flow determines whether heat is absorbed or emitted, *i.e.*, whether temperature is lowered or raised		
4—Thomson	Heat absorption or emission along a single electrical conductor having a finite temperature gradient	Electric current passed through conductor, while temperature gradient is maintained	Yes
	(Flow same as noted for 3—Peltier effect)		
5—Joule	Conversion of electrical energy into heat	Electrical current passed through conductor	No
6—Conduction	Transport of heat or flow of current between two points in any substance	Difference of temperature or electrical potential	No

naturally or by intention; also from crystal imperfection and energy relations within the material. The common semiconductors are considered to include:

Elements	Ge, Se, Si
Compounds	CdS, CuO, CuS, PbS
Intermetallic compounds	Bi_2Te_3, ZnSb, InAs

Electrical conductivity in semiconductors is attributed to the relatively small number of electrons or the presence of so-called *holes*, whereas thermal conductivity is contributed mainly by vibrations of the crystal lattice. Because substances of

high atomic weight have better ratio of electron mobility to heat conductivity, than do those of smaller numbers in the periodic table, they show the best promise.

With semiconductors both thermoelectric power and electrical conductivity depend upon concentration of the current carriers they contain. These carriers comprise both an electronic component and a lattice or structure component. Thermal conductivity may be nearly independent of the carrier concentration when the major contribution to conduction of heat comes from the elastic vibrations of the lattice, as is the case with germanium, a metal that has become prominent recently through its use in transistors which are supplanting vacuum tubes for many types of electronic applications.

With electronic cooling, it is customary to use the term Coefficient of Performance (COP) for expressing efficiency. This is the ratio of heat quantity absorbed (refrigerating effect) to the power required or the *work* input, measured in the same units. An alternate expression is:

$$\text{COP} = \text{heat flow rate/power input}$$

For an ideal thermodynamic device, theoretical efficiency is proportional to the ratio of drop in temperature from source to sink, to absolute temperature of the

TABLE 3—CRITERIA FOR SELECTION OF MATERIAL

PROPERTY	REQUIREMENT
(*a*) Thermoelectric power of junction	High
(*b*) Thermal conductivity between junctions	Low
(*c*) Electrical conductivity of circuit	High

source. Since COP by definition is the reciprocal of heat-power cycle efficiency, it may be expressed as:

$$\text{COP} = T_2/(T_1 - T_2),$$

where

T_1 = temperature of the sink.
T_2 = that of the source.

Values of COP reported to have been attained have been low—under about 0.3 for a temperature range 20 C (36 F). They decrease markedly when the range needed becomes greater.

Mathematical analysis of the principles underlying the thermoelectric effects listed in Table 2 is built upon the well-known three laws of thermodynamics which correlate heat, work and entropy changes; thermoelectric power and temperature are also involved. Such analysis leads directly to the conclusion that there is no theoretical limit on the value of COP that might ultimately be realized. Only practical limitations need be overcome. Moreover, investigations in the huge new field of transistor technology and the advances in so-called *solid state physics* will continue to provide basic knowledge applicable to electronic cooling.

A concrete achievement already attained is the prototype for a household refrigerator announced late in January 1955, the outcome of development work begun in 1951. The demonstration model is described as having a large number of

thermojunctions arranged in 2 groups, for the main cooling compartment and the freezing unit respectively, with heat removal by tap water. The statement was made by the manufacturer, that prior to commencing work 4 years earlier, the best performance known to have been achieved with the Peltier effect was a lowering of temperature by 9 C (16 F). New alloys more recently developed are giving temperature drops several times as great. Work in the laboratories is continuing on the refrigerator and on compact units intended for comfort air conditioning, that will require no external circulating water for heat removal.

In England encouraging results have been reported with junctions using the n-type semiconductor Bi_2Te_3 and metallic Bi which produces 26 C (47 F) maximum temperature difference with coefficient of performance believed as high as 0.25. There are hopes that the electrical-to-thermal conductivity ratio can be further improved by a factor of 4 or 5, and if this is ultimately accomplished, a temperature difference of better than 80 C (144 F) should be realized.

In the United States, thermoelectric effects are known to be under investigation at Battelle, Cornell, Franklin Institute and other laboratories, and among European

TABLE 4—PERFORMANCE CHARACTERISTICS OF PURE METAL AND SEMICONDUCTORS

	THERMOELECTRIC POWER (*a*)	CONDUCTIVITY RATIO (*b*): (*c*)
PROPERTY REQUIREMENT	HIGH	LOW
Characteristic of materials		
Pure metals	Very low	Constant
Alloys	...Wide Range...	
Semiconductors	Quite high	Much lower than for metals

countries in England, France, Germany and Sweden. The work, conducted mainly under the direction of solid state physics specialists, embraces both theoretical and experimental research to determine the materials, geometry and fabrication methods for commercial applications of Peltier junctions, and related factors in thermoelectric behavior.

When solutions to the interrelated problems are found, the principles of *electronic cooling* give promise that they can with equal success be applied to the direct conversion of heat into electric power. Moreover, they may also be found applicable to power generation with radiation from the sun and from the radioactivity that can be created by nuclear reactions.

SOLAR ENERGY

In prehistoric times, man had at his command but 2 primary sources of energy —the sun and the force of gravity. Winds and falling water were harnessed too, but both of these are merely indirect manifestations of solar energy. With the invention of methods for kindling a fire, primitive man obtained a controllable and abundant source of energy. The later discovery of the fossil fuels—coal, oil, gas—and of efficient methods to recover and utilize them, provided world energy resources that until recently have seemed practically inexhaustible. Today, the end

of the reserves that may be exploited commercially is coming into sight. Nuclear fuels will certainly help posterity if not the generation now living, but solar energy seems assured as long as life continues on this planet.

The laws that govern the quantity and angular direction of solar radiation, received at a given spot on the earth's surface, have become well known. Weather Bureau stations, observatories and solar research laboratories are supplying voluminous local data on sunshine availability and its relation to the local needs for heat. The work by R. C. Jordan and J. L. Threlkeld at University of Minnesota, presented recently in a series of ASHAE papers, provides a sound basis for design of utilization systems and equipment. Further useful information is contained elsewhere in ASHAE literature, much of it having originally been secured from the *liability* standpoint, since for summer air conditioning, solar radiation is unwanted and forms a substantial component of the cooling load to be handled by air-conditioning systems.

The primary application for solar energy, that is likely to become useful and economic in the not distant future, is in space heating. Many ingenious arrangements of structure to derive benefit from direct sunshine were devised by the ancients. Further adaptations suited for modern living requirements and architecture are being invented and put to use today. The chief deterrents are economic, since without an effective method to accomplish thermal storage, a supplementary heating source is necessary for use through periods devoid of sunshine, and this is expensive. Seldom can solar energy be applied economically as a supplement to conventional heat sources, merely to reduce fuel bills.

The indirect method of space heating, which involves both a solar collector and a heat-transfer medium, represents the direction of main development effort today. Collector systems are of 3 basic types:

(1) Focusing reflector, essentially a movable mirror.

(2) Flat plate absorber, essentially a shallow insulated box with glass or plastic cover plates.

(3) Intermittent system, employing gases that absorb radiant energy.

In the first type (1), the mirror reflectors must have parabolic shape, either circular or cylindrical, and be moved manually or by motor drive to keep pace with *travel* of the sun. Disadvantages are high initial cost and maintenance expense, along with vulnerability to storm damage, these outweighing the benefits from higher temperature obtained. The second type (2), is based on the *greenhouse* principle, using a box or frame that has at the bottom a dull black plate conductively bonded to a tube through which a heat-transfer fluid is continuously circulated, or an equivalent form of construction if air or a gas is the heat-transfer medium. Four sheets of cover glass give the best thermal performance, but at present prices 2 or at most 3 are the economic maximum. The intermittent principle in (3), whereby gases change their temperature, pressure or molecular form under the effects of solar radiation, remains comparatively undeveloped.

Collector efficiency for the flat-plate type (2) depends on the specific design, materials used including thermal insulation, physical condition, orientation, weather conditions, and upon temperature difference established between the collector plate and outdoors. Some representative values for vertical south-facing collectors having 2 sheets of glass are shown in Table 5. The low efficiencies realized with high collector temperatures on cloudy days require that, for house heating, only moderate temperatures be employed.

In the southeastern region of the United States, the 1000 sq ft of ground area occupied by an average residence receives on a sunny mid-winter day about 500,000 kilo-calories of heat. This is equivalent to 600 kwhr and represents 170 Btu per sq ft averaged over the 24-hr daily cycle. If collector efficiency is 30 percent. the amount useful for heating becomes 180 kwhr, which is about equal to the daily heat requirement. However, some method for heat storage is essential in nearly all cases to maintain acceptable indoor temperature throughout the night hours and on cloudy days.

Thermal storage in many forms has been investigated for dwellings heated by solar radiation and by electricity. The systems tried include:

Massive wall and floor construction, usually of masonry.

Bins filled with rock or gravel, arranged with passageways or pipes for circulating air or water.

TABLE 5—REPRESENTATIVE VALUES FOR VERTICAL SOUTH-FACING COLLECTORS HAVING 2 SHEETS OF GLASS

WEATHER	TIME		BTU PER SQ FT PER HR	EFFICIENCY, PERCENT		
				TEMPERATURE DIFFERENCE, F		
	A.M.	P.M.		50	100	150
Clear	10	2	300	60	48	31
Few clouds	10	2	200	54	36	12
Overcast	Early a.m. to late p.m.		100	37	0	—

Tanks containing heated water or other fluid, with provision for fluid circulation by pumping or thermosyphon effect.

Chemical transition compounds employing the heat-of-fusion principle.

Performance obtained with experimental storage installations has been recorded in many tests and analyzed to determine the best combination of storage type and capacity, and especially to secure correlation with the collector, since the use of storage usually requires higher operating temperature and thus penalizes collector capacity and efficiency. The chemical methods have shown considerable long-range promise, but progress toward a reliable and economic system has been slow. The unsolved problems relate to nucleation, rate of crystallization and solution, vulnerability to corrosion, cost of construction and expense for maintenance.

Another form of thermal storage, applicable to residential heating and universally available, is use of the ground beneath or adjacent to a dwelling. Thermal properties of soils have been investigated through the past decade, to a limited extent with storage of heat gathered by solar collectors as the objective and in greater detail as the means for dissipating heat given off by buried electric cables and as a potential heat source for electric heat-pump systems. Heat extracted from the ground in the latter case is basically derived from solar radiation, either directly or in an indirect manner by transfer from the ambient air and from rainfall. Extensive bibliographies now available cover theoretical analyses of heat transfer

phenomena involved, and the test results obtained from research projects and trial installations of many varieties under the range of working conditions encountered in normally occupied dwellings.

Another method of gathering solar energy is applied for house heating in connection with electric heat pumps that use outdoor air as the heat source. By placing the evaporator coil of the refrigeration (heat-pump) system in a location where it receives incident radiation directly, the heat so absorbed raises the input temperature and improves the heat-pump COP. Basic principles, test results and economic evaluations in this field have been covered in the literature, notably from the work reported by Ambrose, Jordan, Lof, Threlkeld, Voegli and others.

Use of solar radiation as an energy source for summer air conditioning is contemplated in various studies and experimental projects concerned with residential heating. The absorption cycle, adapted to operate with a modest spread between heat intake and rejection temperatures, is employed in the majority of cases.

Solar water heating for domestic uses has a long history in both tropical and temperate zones throughout the world. Design data today are well established. The difficulty of obtaining an economic balance in relation to storage capacity and to supplementary heat from fuel or electric energy during sunless periods and in cold weather, remains the chief restraint to wider adoption of this water-heating method.

Power from solar radiation, both mechanical and electrical, has been an inventor's dream and a research-and-development goal for at least a century. Many ideas have been proposed, condemned, tried, and catalogued. The so-called *solar battery* represents perhaps the greatest advance within the last decade, with the underlying research quite fully disclosed. It utilizes transistor elements to convert the radiant energy from the sun into electricity. Small size units, low efficiency of solar energy conversion, and high costs are still serious limitations, which the sponsors of the solar battery development have freely stated will continue to require talent, money and time.

The present status of solar energy utilization was reported in great detail during November 1955, at the World Symposium on Applied Solar Energy held in Phoenix and Tucson, Ariz., under direction of the *Association for Applied Solar Energy* founded at Phoenix earlier in that year. The extensive proceedings of the Symposium, published several months ago, are commended to engineers and research workers who may become concerned with either the theoretical or practical aspects of solar-utilization technology. The same organization has published a comprehensive bibliography of the literature, extending through nearly a century.

NUCLEAR ENERGY

Among the investigations being conducted under governmental auspices and by private agencies, into the peaceful uses of atomic energy, none has yet given specific attention to residential air conditioning. The so-called McKinney Panel, which issued its report in January of 1956, had received the assignment "to appraise the present and future impact of all aspects of the development of atomic energy on our way of life, our economy, our industry and our natural resources . . . . ." The report makes mention of district heating service but stops short of residential applications. Accordingly, the newly formed Nuclear Energy Engineering Committee of the ASHAE, when considering the scope of activities it might usefully undertake, will doubtless have that subject on its agenda.

Principal directions in which future developments in the fields of atomic energy might ultimately affect residential air conditioning are:

1. As a supplementary *fuel* or heat energy source for bulk generation of electric power, thereby tending to extend the useful life of present conventional domestic fuels.
2. As assurance that electric power will continue to be available, abundantly and at low cost.
3. Under circumstances favorable to the establishment of district heating systems, as a fuel for producing steam or hot water for distribution to consumers.

In each of these cases, the utilization of nuclear fuels would be confined to relatively large installations—of capacity perhaps ranging between 100,000 and 1,000,000 lb steam per hr, and under conditions permitting operation at a high annual load factor. Smaller installations and low load factors would be penalized by excessive cost and space requirements for the bulky shielding to confine the dangerous radiations, and for other safety measures.

The so-called *nuclear fuels*, which release heat energy through fission in a controlled chain reaction, are the heavy elements uranium and thorium. Estimated world reserves of uranium reported a half-decade ago had between 20 and 25 times the total heat energy in the available fossil fuels—coal, oil, gas—recoverable commercially at cost levels not exceeding twice the current costs; thorium was considered to be more plentiful than uranium. Today, the estimated reserves of uranium have been multiplied by a factor of 2 or 3, as the result of continuing explorations with more refined prospecting techniques and devices, and more adequate knowledge of the geologic formations in which uranium can occur.

Thermonuclear fuels, which operate on the fusion principle and include certain forms of hydrogen, lithium and perhaps ultimately other light elements, have greatly augmented our potential energy reserves and initially our stockpiles of weapons. These fuels can be expected sooner or later to become usable in controlled reactions, releasing heat for thermal applications and for electric power generation, as well as for propulsion, in a manner comparable to that employed with the fission process. The outlook for utilizing the fusion process is uncertain today, owing in part to delays by governmental agencies in releasing to industry the needed basic information and the right to investigate nonmilitary applications of fusion technology.

The methods presently contemplated for utilizing nuclear fuels all involve use of some type of reactor—an apparatus in which a sufficient quantity of uranium or thorium is placed in such manner that a controlled chain reaction can occur, the released thermal energy then being transferred to associated apparatus of conventional types. Many forms of reactors are under investigation for generation of electric power and for propulsion of ships and aircraft. The same types will likewise become applicable to the production of heat for warming buildings, subject to the specific economic and practical considerations that pertain to space heating. Since comprehensive reviews of the present status and the outlook for civilian atomic power have been appearing in current publications, including notably the May 28, 1956 issue of *Electrical World* magazine, the presentation here does not undertake to go into the subject.

The possibility exists of obtaining atomic power from radioactive substances, by utilizing the properties of semiconductors. A single high-energy electron emitted from a radioactive atom can produce 200,000 slower electrons in the bombarded semiconductor, thereby establishing an electric current. The so-called *atomic battery* announced last year is an example. Although still in the research-and-

development stage, with power measured only in millionths of a watt, progress is continuing toward units of greater capacity. Initial applications contemplated are for miniature devices like hearing aids, pocket-size radio receivers, signal controls and similar devices for which extreme compactness, reliability and great length of service are important. Conceivably the atomic battery, and the solar battery as well, might some day be combined with electronic refrigeration for comfort air conditioning.

Two other aspects of nuclear technology may be mentioned. Proposals to use radioactive fission products, recoverable from spent fuels after removal from nuclear reactors, received publicity a year or two ago as a potential heat source for residential heating. Also, there has been speculation on the possibilities for direct production of electric power from energy contained in the atom, without passing through a conventional heat-power cycle. Both these subjects are reviewed in the mentioned report of the McKinney panel. In brief, they are considered to hold little promise for tangible applications to residential air conditioning within the foreseeable future.

CONCLUSION

The outlook for continued expansion and diversification of residential air conditioning was never brighter. Electronic cooling is but one of many avenues being explored that will lead to novel systems and to improved components and combinations for systems currently in use. Energy resources will surely be augmented by the expanding research programs and by the developments aimed at making solar radiation and nuclear energy universally available. However, economic factors will require as careful attention as the technical problems involved.

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HONORARY MEMBERS—Elected 1956

HOOVER KETTERING EISENHOWER

HERBERT CLARK HOOVER

Herbert Hoover was born August 10, 1874 in West Branch, Iowa.

His father and mother died at an early age, and therefore on entering College at 17 years of age it was necessary for him to provide all finances for his education at Leland-Stanford from which he was graduated in May 1895 with a degree in Engineering.

For many years he followed the profession of a mining engineer.

In 1900 following the Boxer Rebellion in China he was given a position of Chief Engineer for a Reorganized Mining Company. This company was later taken over by Belgian interests.

In 1908 Hoover severed all connections with the mining company and established a consulting firm—as by now he was foremost in the field of Mining Engineering. By 1914 he was Managing Director or Chief Consulting Engineer for more than a score of mining companies throughout the country.

In the spring of 1914 he was in London to arrange for the participation in the Panama Pacific Exposition of 1915 which lead to his position as Chairman of the Commission for Relief in Belgium, 1915–1919—during which time he fed ten million people.

After this outstanding work in Europe, Hoover retired to the United States and accepted the post of Chairman of the Supreme Economic Council and later that of Secretary of Commerce.

In 1928 he was elected President of United States and served for a term of four years.

Since that time Mr. Hoover has written many outstanding books and has given advice to America on many other problems. His most recent public activity is his service as Chairman of the Commission on Organization of Executive Branch, U. S. Government.

In recognition of his outstanding career, Mr. Hoover has been the recipient of degrees from many American and foreign universities. He has also received many medals and degrees from American institutions as well as from those in other parts of the world.

CHARLES FRANKLIN KETTERING

Charles F. Kettering was born on a farm near Loudonville, Ohio, August 26, 1876. He attended Loudonville High School and Wooster Normal School.

At age 19 he became a country schoolteacher at Bunker Hill, Ohio which he left a year later intending to attend Ohio State University. Due to an eye ailment he was

forced to postpone the studies for two years and then was graduated from Ohio State in 1904, at the age of 28 with degrees in Mechanical and Electrical Engineering.

Mr. Kettering first went to work for the National Cash Register Company and became chief of inventions department. He resigned in 1909 and formed the Dayton Engineering Laboratories with Mr. E. A. Reeds former works manager for National Cash Register.

During the next two years, Kettering improved automobile ignition and lighting systems and invented the first self-starter for automobiles.

In 1916 Delco was purchased by General Motors formerly United Motors Corporation and in 1925 Kettering joined General Motors as head of its Research Laboratories which were moved to Detroit, and as President and General Manager of the Division, he was responsible for many developments. Among these were quick drying lacquer, anti-knock gasoline and the development of a high-speed two-cycle engine.

Since 1928 he has received a number of honorary degrees including Doctor of Science degrees from sixteen universities, four Doctor of Laws degrees, four engineering degrees and also the Washington award, the Franklin Gold Medal as well as many other awards and citations for outstanding achievement.

Kettering was President of the *Society of Automotive Engineers* in 1918 and of the *American Association for the Advancement of Science* in 1945. He is also a member of numerous organizations including many in engineering, such as: *ASME, AIEE, ASTM,* and *Amer. Soc. of Civil Engineers.* Mr. Kettering's interest in the research and development phases of engineering has been lifelong.

MILTON STOVER EISENHOWER

Milton S. Eisenhower was born in Abilene, Kansas on September 15, 1899. He obtained his college education at Kansas State College, receiving a B. S. degree in 1924, and has since been the recipient of 14 honorary degrees from various universities.

Dr. Eisenhower was engaged in consular work abroad from 1924–1926 and in Washington was with the U. S. Department of Agriculture from 1926 to 1942. During World War II, he served as Director of the War Relocation Authority, March-June 1942 and as Assoc. Director Office of War Information, June 1942–1943.

In 1943, Dr. Eisenhower became President of Kansas State College continuing until 1950 when he became President of The Pennsylvania State University.

He has long taken an active part in the work of many organizations including serving on the Special Committee on Government Organization since 1953; Chairman, U. S. National Committee for UNESCO, 1946–1948; Delegate UNESCO Conference 1946, 1947, 1948, 1949. Member of President's Commission on Higher Education, 1946; Problems and Policies Committee of American Council on Education since 1950; Executive Committee Association of Land-Grant Colleges and Universities, 1944–1947, 1950–1953 Chairman 1946–1947, 1952–1953 President of the Association, 1951–1952; Chairman of National Committee for The People Act, 1951–1953; Director Fund for Adult Education since 1953; Trustee National Committee for Economic Development, 1947–1951; National Society for Crippled Children, 1950–1953; Institute for Intl. Education, 1951–1953; Director of Freedoms Foundation Inc., since 1951, Chairman General Awards Jury, 1950; Director The Geisinger Memorial Hospital since 1952; Chairman American Korean Foundation since 1952.

Dr. Eisenhower was the recipient of the Horatio Alger Award for 1952. He is a member of Phi Kappa Phi, Sigma Alpha Epsilon, Sigma Delta Chi, Alpha Zeta. He has contributed numerous articles to popular publications and is the author of numerous federal bulletins and leaflets. Always interested in engineers and engineering education, he was the principal speaker at the 1955 Annual Banquet of the Society in Philadelphia.

In Memoriam 1956

Name	Joined
Herbert G. Angelo, Denver, Colo.	1944
William V. Bibb,* Ottumwa, Ia.	1950
Robert C. Clarkson, Jr., Philadelphia, Pa.	1921
William T. Conroy, Kansas City, Mo.	1944
John T. Davis, Houston, Tex.	1945
Charles F. Donohoe, Detroit, Mich.	1941
George L. Erwin, Jr., Chicago, Ill.	1954
Ralph S. Franklin (*Life Member*), Boston, Mass.	1919
John Frei, Berne, Switzerland	1951
Frank L. Gray, Jr., Dallas, Tex.	1945
Edgar Herring (*Life Member*), London, England	1919
Charles A. Hoppin, Sr., St. Louis, Mo.	1946
John C. Hornung (*Life Member*), Tucson, Ariz.	1914
Thomas B. Hunter,* San Francisco, Calif.	1941
Lynn H. Jackett, Cleveland, Ohio	1946
Elmer S. Jaggars, El Paso, Tex.	1955
Frank P. Keeney (*Life Member*), Chicago, Ill.	1915
Norton W. Kingsland, Toronto, Ont., Canada	1945
Frank E. P. Klages, Greensboro, N. C.	1940
Robert E. Kramig, Jr., Cincinnati, Ohio	1933
Herbert Lemkau, San Francisco, Calif.	1946
J. R. Lucien L'Esperance,* Montreal, Que., Canada	1948
Nicholas J. Mathey (*Life Member*), St. Joseph, Mo.	1915
Ambrose A. Maycock, Salt Lake City, Utah	1945

* Died in 1955 but Society not notified until 1956.

In Memoriam 1956

(*continued*)

NAME	JOINED
Ernest W. McMullen, Boston, Mass.	1942
Herman W. Nelson (*Life Member*), Moline, Ill.	1909
Heinrich M. Nobis (*Life Member*), Lakewood, Ohio	1914
Charles A. Potteiger, Reading, Pa.	1947
Orville Pratt, Omaha, Nebr.	1953
Carlton P. Roberts, New York, N. Y.	1952
Robert M. Rosebrough (*Life Member*), Webster Groves, Mo.	1920
John D. Ross,* Montreal, Que., Canada	1937
William H. Severns, Urbana, Ill.	1933
Douglas W. Shand, Montreal, Que., Canada	1952
Vincent E. Small, Pasco, Wash.	1956
Benjamin H. Snow, Melrose, Mass.	1951
William Stein, New York, N. Y.	1946
Stephen G. Swisher, Jr., Milwaukee, Wis.	1936
Ernest Szekely, (*Life Member and Presidential Member*), Milwaukee, Wis.	1920
Charles E. Towner, Chicago, Ill.	1950
James V. Ursini, New Haven, Conn.	1948
Elmer A. Whitelaw, Chicago, Ill.	1950
Herman J. Woehlke, Denver, Colo.	1944
Charles A. Wood, Dayton, Ohio	1937
Baldwin M. Woods (*Presidential Member*), Berkeley, Calif.	1937
Hans J. Zack, Chicago, Illinois	1928

* Died in 1955 but Society not notified until 1956.

Ernest Szekely

1888-1956

Ernest Szekely, president of the Society in 1952 and president of Bayley Blower Co., Milwaukee, Wis., died at his home in Wauwatosa on Thursday, November 29, 1956, after a prolonged illness.

Born July 24, 1888 in Debrecen, Hungary, and graduated from the University of Budapest in 1910 with an M.E. degree, Mr. Szekely began his engineering career as assistant and later became associate professor at the University.

After coming to the United States, he worked as a designer for Guarantee Construction Co., New York, and as a research engineer for American Blower Co. He had a consulting engineering office in Cleveland for many years, specializing in public buildings and schools. Later, he became chief engineer for Drying Systems, Chicago, and then chief engineer for Bayley Blower. The climax of his career came in 1932 when he was named Bayley's president.

Mr. Szekely, who joined the Society in 1920 and became a Life Member in 1954, was extremely active in both the Northern Ohio and Wisconsin Chapters, as well as in Society activities. He served as a member of the Council, 1945–47, and on numerous Society committees, including the Nominating Committee and the Code Committee for Exhaust Systems. In 1949, he was treasurer of the Society and a member of the Finance Committee of the Council, and in 1950 was chairman of the Finance Committee. The following year he became first vice president and chairman of both the Executive and the F. Paul Anderson Committees.

In 1952 Mr. Szekely served as Council chairman, and in 1953 as a member of Council and chairman of its Executive Committee, as well as chairman of the Advisory Board. Since 1954 he was a member of the Advisory Board.

In addition, Mr. Szekely served on many Technical Advisory Committees, including the TAC on Infiltration in Buildings; the TAC on Effect of Entering Temperature and Velocity on the Temperature and Distribution of Air Within an Enclosure, serving as its chairman in 1936; and the TAC on Air Distribution, 1937 to 1950, serving as its chairman in 1937.

Mr. Szekely served the Northern Ohio Chapter as vice president in 1923 and president in 1924, and the Wisconsin Chapter as second vice president in 1933, first vice president in 1934 and president in 1935. From 1936 to 1938 he was a member of the board of governors of the Wisconsin Chapter and also had served on many of the chapter's code and other committees.

Mr. Szekely wrote numerous engineering articles and is listed in *Who's Who in Engineering*. He was a member of the *ASME*, the *Wisconsin Society of Professional*

Engineers, the *Astronomical Society*, and president of the *Engineers Society of Milwaukee* 1946–47. He also held membership in Rotary International, Tau Beta Pi and Pi Tau Sigma, and Marquette University faculty. He was a Presbyter in the Evangelical Reformed Church, and director of Town Hall of Milwaukee.

Mr. Szekely was an accomplished musician and had travelled widely. He had a host of friends who enjoyed his fine sense of humor. Many of the Council Meetings were enlivened by his witty observations.

He is survived by his wife, Sari, his son George E. Szekely, and two grandchildren.

Services were held from the Ritter Funeral Home on Monday, December 3, at 2:00 p.m.

Baldwin M. Woods

1887-1956

The Society received, with very deep regret, news of the untimely death on Friday, September 7, at the age of 68, of Dr. Baldwin M. Woods.

Dr. Woods was president of ASHAE in 1947, an office in which he served with great diligence and distinction, and later as a presidential member he was actively and usefully engaged in Society affairs, remaining on the advisory board until his death. He had retired last year as vice president, university extension department, University of California, where he had been on the faculty for 47 years. He will be greatly missed by his many friends.

Born in Lampasas, Tex., on September 22, 1887, he received his degree in electrical engineering from the University of Texas in 1908. He received his M.S. from the University of California in 1909 and Doctor of Philosophy from the same university in 1912. Continuing his studies in Europe, Dr. Woods was a student at the Université de Paris in the fall and winter of 1912 and 1913, and also attended the summer semester at the Universitat zu Munchen in 1913.

Initiating his professional career in 1907 with an assistantship in applied mathematics at the University of Texas, he later went to the University of California as a John W. Mackey, Jr., fellow in electrical engineering in 1908, and served as instructor in mathematics, 1910–14. During the years 1914–15 he was instructor in theoretical mechanics, and was assistant professor of theoretical mechanics, 1915–19. He was appointed professor of aero-dynamics in 1919 and occupied that position until 1930 when he became professor of mechanical engineering.

Widely experienced as an administrator, Dr. Woods was dean of the summer session in Los Angeles at the University of California, 1921–22 and during 1923–25 he served as assistant dean of the university. He was appointed associate dean in 1924, was representative in educational relations 1923–30, which latter year saw him appointed to the chairmanship of the mechanical engineering department. He became director of the university extension department in 1942 and in 1950 was named vice president, university extension, retiring in 1955.

During the first World War, Dr. Woods was president of the academic board and director, U. S. School of Military Aeronautics, University of California, 1917 to 1919, and in the second World War was vice chairman, University War Council, from 1942 to 1944.

In addition to his extensive professional and administrative duties, Dr. Woods undertook heavy responsibilities in local and national affairs. He served as consultant in a Survey of Federal Research in the United States with the National

Resources Committee in Washington from 1937 until 1938. He maintained relations with the National Resources Planning Board upon his return to the university, serving as chairman of the region covering California, Arizona, Nevada and Utah until 1943. He was chairman of the Water Resources Committee until 1943.

Dr. Woods was appointed federal coordinator, California State Council of Defense, in 1940. He was also chairman of the advisory board for aeronautics for the University of Illinois during 1944–46.

Thoroughly experienced in public relations, Dr. Woods was chairman of the National Committee for the Award of Boeing Scholarships 1930–38, and chairman, National Committee for the Award of United Air Lines Scholarships, 1939–42. He served on many *ASME* and *SPEE* special committees and on the board of awards for the Daniel Guggenheim Medal Fund, Inc., from 1934 until 1937.

He was the author, in collaboration with Frederick E. Pernot, of a book on logarithms of hyperbolic functions to twelve significant figures, 1916; *Dynamics of Aeroplanes* (with John E. Younger) in 1931; and *Introduction to Engineering Economy* (with E. Paul DeGarmo) in 1942. He also contributed technical and nontechnical articles to various publications.

A fellow of the *American Association for the Advancement of Science*, Dr. Woods was also a member of *ASME*, *SPEE*, and *ASEE*, an honorary member of the *Air Conditioning Society of San Francisco*, a founder member of the *Institute of the Aeronautical Sciences*, a member of the *Engineers Council for Professional Development*, *National Planning Association*, *National University Extension Association*, *Adult Education Association*, *American Association of University Professors and the U. S. Naval Institute*. He was a member of Sigma Xi, and president of its California chapter in 1931, a member of Tau Beta Pi, Alpha Chi Rho, and Phi Delta Kappa, and an associate member of Eta Kappa Nu.

Joining ASHAE in 1937, Dr. Woods became the first president of the Golden Gate chapter, one which he was instrumental in organizing. He was soon active on chapter and Society committees and became a member of Council in 1942. Later, as Society First Vice President, he served as chairman of the F. Paul Anderson Award Committee.

In the course of his long and energetic career he served the Society in the following capacities:

1938	Speakers Bureau
1938–1939	Engineering in its Relation to Public Health
1938–1940	Committee on Research
1939	Committee on Engineering Scholarships
1940–1941	Advisory Committee for Pacific Heating and Air Conditioning Exposition
1940	Executive Committee, Committee on Research
1940–1942	TAC on Cooling Towers, Evaporative Condensers and Spray Ponds (chairman)
1942–1944	Member of Council
1942	Membership Committee
1943	War Service Committee
1944–1946	Finance (chairman)
1945	Research Fund Raising
1946	F. Paul Anderson Award (chairman)
1945	Second Vice President
1946	First Vice President
1947	Executive (chairman)
1948	Advisory Council (chairman)
1948	Public Relations
1949–1956	Advisory Board
1949	Federal Liaison
1950–1952	International Joint Committee on Psychrometric Data (chairman)

Services were held on Monday, September 10, at the First Methodist Church, Oakland. Honorary pall bearers were Admiral Chester Nimitz, Mark Davids, Thomas N. Barrows, Paul H. Sheats, Roy Heffner, and L. M. K. Boelter. The In Memoriam address was given by Dr. Robert G. Sproul, president, University of California.

Dr. Woods is survived by his wife, Elizabeth; three sons, Baldwin, Robert and Ronald; two sisters, Josephine and Eloise; and three grandchildren; to whom the members, the Officers and the Council extend their sympathy.

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